Power and The Engineer

Vol. 32 NEW YORK, JULY 5, 1910 No. 27

THERE has been a great deal of discussion regarding the exceedingly small remuneration paid to operating engineers, and the subject deserves every attention that can be given to it. There is still another important member of the operative staff who should not be overlooked —the switchboard attendant.

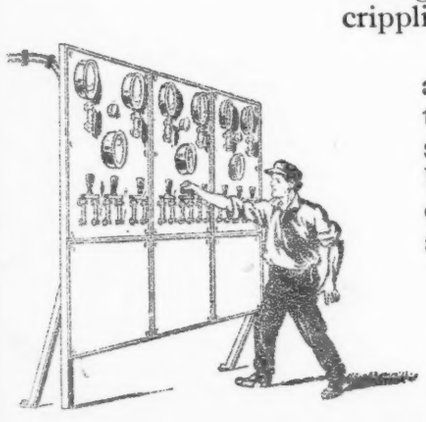
While a good operating engineer may cut a large slice from the coal bill, an inefficient switchboard attendant may add an equally large amount to the repair bill.

To be a good switchboard operator, a man should have had not only experience, but should also have acquired a certain amount of knowledge regarding the theory of electricity.

In the steam end of the plant, when anything goes wrong, it usually happens in plain sight or manifests itself in such a manner as to be easily traceable; but in the electrical end only the effect is evident at the switchboard, and it is up to the attendant to solve the cause and to decide almost instantaneously upon his course of action.

He has to use his brains and use them quickly.

An error in judgment at the critical moment, or the chance pulling of the wrong switch may mean hundreds of dollars of damage, as well as crippling the service.



A person with an excitable temperament should never be allowed to operate a switchboad.

It requires men of intelligence and sound judgment.

The question arises, can men possessing these requisites be obtained at the meager rate of wages paid for this class of service?

In New York City the average rate for a switchboard attendant is $3 per day, while a union electrician receives $4.50 per day. Without going into the respective requirements of the two—which would certainly be in favor of the former—and neglecting the question of continuous employment, let us confine ourselves to a consideration of the relative responsibilities.

The electrician is responsible only for the particular piece of apparatus upon which he is working. This is usually not in service, and before being put into use is inspected by the foreman.

On the other hand, the switchboard attendant may have the responsibility of a large part of the electrical service resting upon his shoulders.

Owing to the fact that the development in switchboard work has been so rapid, there is a younger class of men engaged in this line than in the steam end of the plant. Some of these men are highly capable, but the compensation is not attractive enough to hold them for any length of time, and they seek other fields of advancement.

The frequent breaking in of new men is not conducive to efficient operation and should not be tolerated for the sake of a small reduction in operating expenses.

Brains are needed in this position, and to secure intelligent men and retain them for any length of time demands a fair return for their services.

Hydraulic Turbines for 670-Foot Head

By H. Birchard Taylor

A distinct advance in the design and application of hydraulic turbines is marked by the construction of two 6000-horsepower reaction turbines which are to be operated under a 670-foot head at Noriega, Mexico. Owing to this high head, special attention had to be given to the proportioning of parts and to the means of regulation.

There are now in the course of construction, in the shops of the I. P. Morris Company, of Philadelphia, two hydraulic reaction turbines which will be applied to a head higher than has ever before been attempted for this type of water-wheel. They are to be installed in the power plant of the Michoacan Power Company at Noriega, Mexico.

The turbines are of the horizontal-shaft volute-casing, inward-flow, Francis type and each is designed to deliver at its flange coupling, 6000 horsepower, when operating under an effective head of 670 feet at a speed of 514 revolutions per minute. They will be directly connected to 3500-kilowatt, 60-cycle, three-phase generators, manufactured by the General Electric Company.

When delivering 6000 horsepower under the above conditions of head and speed, each turbine will require 98.7 cubic feet of water per second. Therefore, when both units are delivering full power, 197.4 cubic feet of water per second will be required. The water will be supplied to both turbines through a single penstock about 5600 feet in length.

It has been only within the last few years that the reaction type of turbine has been applied to heads much over 300 feet. Some of the more recent applications of the reaction turbine to high heads have proved conclusively that the erosion of the runner vanes in high-head turbines is due primarily to defective design, and not to foreign substances in the water, poor material, electrolysis, or chemical action.

The successful operation of the four 18,000-horsepower turbines in the plant of the Great Western Power Company, on the Feather river, near Oroville, Cal., has greatly strengthened the confidence of engineers in the reaction turbine, as a type well adapted to compete with the impulse turbine, for heads probably as high as 1000 feet. The runners of the Great Western turbines are not all of the same material; two are made of bronze, one of cast steel and one of cast iron. All have been operating continuously for a year without showing any signs of erosion. The present head under which they operate is 420 feet, but this will be increased to 525 feet when the dam is completed.

Hydraulic engineers are awaiting with considerable interest the time when the Noriega turbines will be put into commercial operation, as the construction of these units marks a distinct advance in the art of hydraulic-turbine design and their successful operation will greatly broaden the field for the reaction type.

The turbines are of the single-runner, single-discharge type with outside operating mechanism. Each casing is made of cast steel in two sections, the division occurring on the horizontal plane which passes through the axis of the shaft. The normal pressure in the casing will be about 285 pounds per square inch, but owing to the wide fluctuations of pressure resulting from the sudden stopping and starting of the water column in the penstock, the turbine casing has been designed to withstand a pressure of 585 pounds per square inch. The sizes of the various parts of the turbine subject to this pressure are so proportioned that the resulting stresses will not exceed the elastic limits of the metals used in their construction.

Each casing is securely tied by 17 heavy forged-steel staybolts, which, aside from preventing excessive strains in the metal of the casing by taking a large proportion of the load upon themselves, are so arranged as to prevent an appreciable change in the width of the throat opening of the casing where the guide vanes are situated. Both casings have been subjected to a hydraulic-pressure test of 585 pounds per square inch. The accompanying illustrations show one of the turbines erected in the shop for this test.

The runner of each turbine is made of special bronze in a single casting and is securely bolted to a hub forged on the shaft. All the vane surfaces have

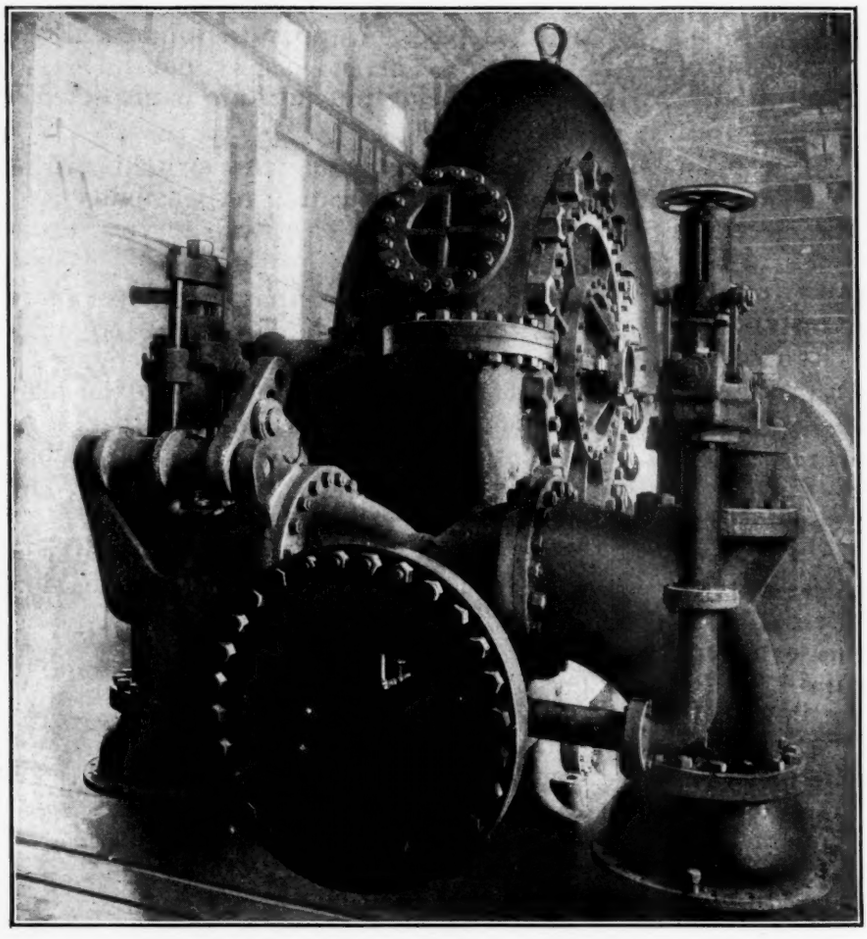


FIG. 1. VIEW SHOWING RELIEF VALVES

been finished smooth so as to reduce to a minimum the frictional losses.

Provision has been made to balance the hydraulic thrust on the runner, but should any unbalanced thrust occur it will be taken care of by a thrust bearing located on the tail end of the shaft.

The forged-steel shaft which revolves in two ring-oiling bearings is made of such a diameter as to prevent any appreciable deflection at the runner. It is therefore possible to use scant clearances between the runner and the stationary parts, thus preventing unnecessary losses in efficiency by cutting down the leakage at the seals.

The water will be distributed to the runner through cast-steel movable guide vanes. Each vane has cast with it a fulcrum pin, which extends from both sides of the vane. The section of the pin on the generator side is short in length and extends into a bearing on the inboard distributer plate. The operating section of the pin, which is long, extends through a stuffing box and two bearings on the outboard headplate. Thus each fulcrum pin is held in alinement by three bearings.

The operating part of each pin is connected by a cast-steel lever and a cast-iron link to the cast-steel operating ring, which turns in a groove in the outboard headplate. The lower part of the ring is connected to the piston rod, which is controlled by the main operating cylinder mounted on the turbine casing.

The entire operating mechanism is so designed that should any two vanes close on a stick which has passed through the racks at the headgates, the two vanes will be capable of sustaining the entire force of the governor, without causing any permanent distortion of the vanes. This arrangement not only reduces the maintenance charges by increasing the life of the guide vanes, but it also prevents excessive leakage when the vanes are in the closed position.

The penstock will terminate in a cast-steel Y-pipe located outside of the power house and each turbine will be connected to a branch of this Y. Between each turbine and the penstock will be located a 36-inch, cast-steel body, bronze-mounted, hydraulically operated Ludlow gate valve. The operating cylinder of each valve is designed to operate the valve when acted upon by a pressure considerably less than that to which the valve itself will be subjected. Each valve is equipped with a 6-inch bypass for equalizing the pressure on both sides, but the cylinder is capable of operating the valve with the pressure on one side of the valve only.

The thrust-bearing pedestal is mounted on a subbase which in turn is mounted on a sole plate securely bolted to the concrete foundations. Thus, if it should be desired to dismantle the turbine, the bolts connecting the cap to the pedestal may be removed and the bolts which connect the pedestal to the sole plate may then be withdrawn. The subbase may then be slipped out from beneath the pedestal and the latter dropped down and removed from under the thrust collars on the shaft. The draft bend and the headplate containing the operating mechanism may then be unbolted and removed from the casing and slipped over the tail end of the shaft.

Owing to the high head under which these turbines will operate and the likelihood of the seals between the runner and the stationary parts increasing at a rapid rate, the turbines have been equipped with removable cast-iron plates at these points. These plates may be removed from time to time when the clearances become too large. Thus no permanent damage to the turbines will result from the action of the water.

Removable plates are also supplied on both sides of the throat of the casing where the guide vanes are situated. These plates are made of cast steel and will protect the turbine casing from wear caused by the scouring action of the water, as it passes with high velocity through the guide vanes.

Each turbine is supplied with a small hand-operated oil pump mounted on the operating cylinder and connected through piping to a jack cylinder superimposed upon the end of the operating cylinder. The piston rod which controls the operating ring passes through the main operating cylinder and terminates at a piston in the jack cylinder. In case the governor system should fail the turbine may be operated by the hand pump through the jack cylinder.

The problem of designing the turbines for good speed regulation has been an extremely difficult one, owing to the fact that they are fed through a penstock 5600 feet long and that the flywheel effect of the generator amounts to only 108,000 pounds at one foot radius.

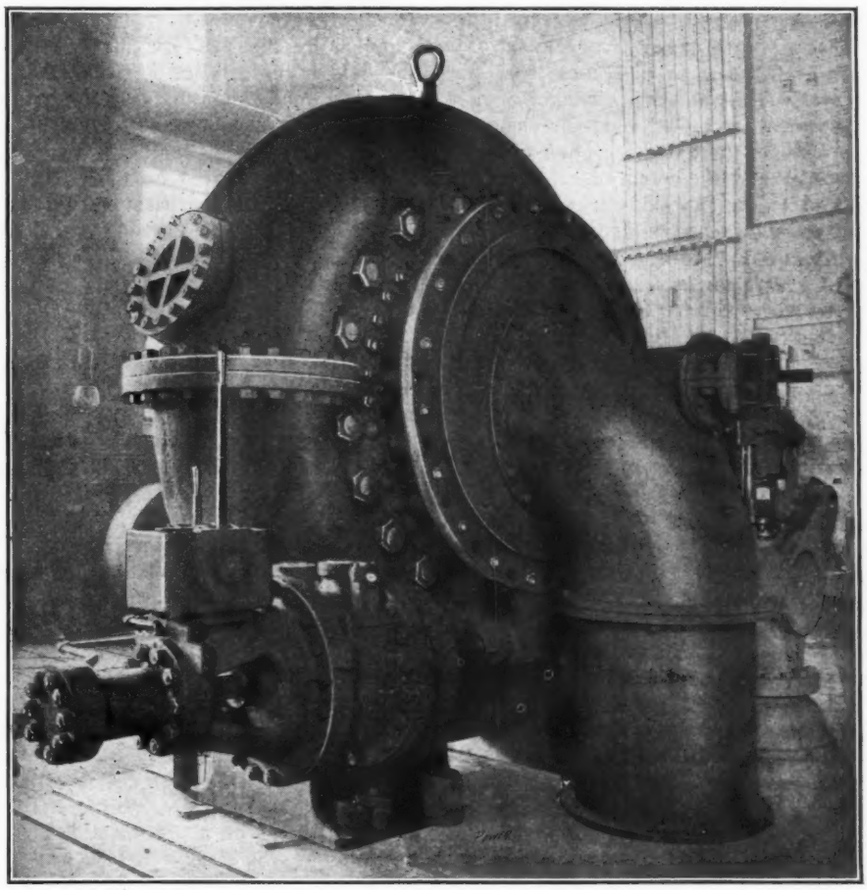


FIG. 2. SHOWING OPERATING MECHANISM

The speed regulation of a turbine is not wholly dependent upon the installation of a first-class governor. Of course, a first-class governor is essential to good governing but the best governor constructed will not insure good regulation if the conditions are not favorable.

The principal factors entering into the regulation problem of every turbine which is supplied with water through a penstock are as follows:

The inertia of the water in the penstock.

The speed with which the turbine gates are opened or closed by the governor to suit the changes in the load.

The flywheel effect of the revolving parts of the turbine and generator.

Turbines which are fed by long penstocks are subject to wide fluctuations of the pressure in the wheel casing. The

magnitude of the pressure change depends upon the length of the penstock and upon the rate of the change in the velocity of the water column in the penstock. The latter depends upon the velocity with which the gates are moved to suit the variations in the load on the turbine.

When the load suddenly drops, the gates are partly closed by the governor and the quantity of water delivered to the runner is thereby reduced to an amount required to balance the new load, at the normal speed. The closing of the gates reduces the velocity of the column of water in the penstock and, as it requires force to change the velocity of a body, this force is evidenced by a rise in pressure in the wheel casing.

Again, when a load suddenly comes on the turbine, the gates are opened by the governor and the quantity of water delivered to the runner is increased to an amount required to balance the new load at the normal speed. The opening of the gates allows an increase in the velocity of the water column in the penstock and as it requires a force to change the velocity of a body, this force is evidenced by a drop in the pressure in the wheel casing.

If the pressure in the wheel casing is increased or decreased, it is equivalent to operating the turbine under a higher or lower head and this change in the head exists during the time in which the gates are being adjusted to suit the change in the load. Should the gates be moved too rapidly, the pressure change will become a greater factor in controlling the speed of the turbine than the change in the quantity of water resulting from the movement of the gates.

To illustrate this point, let us assume that a portion of the load on the turbine drops off and the gates close to reduce the amount of energy supplied to the runner. The rise in the pressure will increase the velocity of the water through the gates and the quantity delivered through smaller gate openings will be considerably above the normal discharge for these openings based on a constant head. The product of the increased pressure times the quantity of water delivered per second is a measure of the energy supplied to the runner and this energy might be greater than that which would have been the case had the pressure remained constant during the closing of the gates. Thus, the whole scheme of regulation is upset.

In practice the pressure fluctuations are usually modified by the installation of standpipes, surge tanks and relief valves, but in all cases the allowable velocity of the gates is largely dependent upon the design of the penstock.

Assuming that the movement of the gates does not cause a change in the pressure in the casing, then if the load goes off and the gates close to suit the new condition of load, the excess energy supplied to the runner during the movement of the gates is taken up by the revolving parts and the speed of the turbine increases. For a given amount of energy the rise in speed depends directly upon the flywheel effect, or the weight multiplied by the square of the radius of gyration of the revolving parts. The greater the flywheel effect, the smaller the speed change.

Conversely, when the load on the turbine is suddenly increased and the gates open to suit the new load, the deficiency in the energy supplied to the turbine, during the movement of the gates, is supplied by the revolving parts as the speed is reduced. For a given amount of energy taken from the revolving parts, the drop in speed depends upon the flywheel effect.

Thus, it is seen that for good governing, it is desirable to have as great a flywheel effect as is practicable and in cases where the revolving parts of the generator will not supply the value sufficient to obtain the degree of regulation required, it is then necessary to increase the inertia of the revolving parts by the addition of a flywheel on the turbine shaft.

The various factors entering into the problems of regulation—the inertia of the water in the penstock, the relief valves, the standpipe, the flywheel effect, and the speed with which the gates are moved, are all more or less related and it is essential that they be carefully studied and each factor given its due attention by the engineer in making his guarantees of regulation.

In order to secure the regulation desired by the Michoacan Power Company, it was found necessary to increase the flywheel effect (Wr^2) of the revolving parts to 282,000 pounds at one-foot radius, by the addition of a cast-steel flywheel weighing approximately 20,000 pounds. It will be located between the generator bearing and the turbine bearing, and will be mounted and keyed to the generator shaft and bolted to the forged-flange coupling on the turbine shaft.

The flywheel is so designed that when operating at the runaway speed of the turbine, which will be about 900 revolutions per minute, the stress in the rim will not exceed the elastic limit of the material.

Owing to the question of safety as well as speed regulation, it is necessary that the variation of the pressure in the turbine casing, caused by the stopping and starting of the water column in the penstock, shall not exceed certain limits. In order to control these variations in pressure, each turbine is equipped with a mechanically operated, cast-steel body, relief valve connected to the turbine casing.

The relief valve is arranged to be actuated by a mechanical connection through a dashpot, to the operating ring of the gate-operating mechanism, in such a manner that, when the turbine gates are suddenly closed, the relief valve will open. The construction of the relief valve is such as to discharge practically the same amount of water that is shut off by the turbine gates.

The relief valve is also arranged so that it may be operated as a synchronous bypass, discharging water in inverse proportion to that being used by the wheel, thus keeping the velocity in the penstock constant and preventing variations of the pressure in the wheel casing. It may also be arranged so as to be used as a water-saving bypass, that is, when the turbine gates are suddenly closed, the relief valve will open and prevent an abnormal rise of pressure, and will then slowly close at such a rate as to keep the pressure in the pipe line within the required limits.

There will also be provided for each turbine and attached to the turbine casing a steel-body, pressure-actuated, relief valve which will be capable of discharging the full quantity of water used by the turbine. This relief valve will be operated by a rise of pressure in the pipe line, and will be so designed as to open in proportion to the rise in pressure and with a decrease in pressure, the valve will close and shut off a proportionate amount of water.

The gates of the turbines will be controlled by Morris governors of the double-floating lever, oil-pressure type. Each governor will be operated by a belt from the turbine shaft, and in addition is equipped with a motor so that it may be controlled from the switchboard. In order that the revolving element of the governor may be stopped during the operation of the turbine without disturbing the driving belt, a friction clutch is provided on the governor pulley.

The governing apparatus of each turbine consists of a governor, accumulator tank, three-throw oil pump and the interconnecting piping. The accumulator tank is divided into pressure and vacuum compartments. The oil pump is of the Deane, vertical, three-throw type and is driven from the turbine shaft by means of a belt. The interconnecting piping and fittings between the pump, governor, accumulator tank and operating cylinder are made of brass and all parts are so designed that the governor will control the turbine gates satisfactorily with a working pressure not exceeding 150 pounds per square inch.

When operating under an effective head of 670 feet at the normal speed of 514 revolutions per minute, each turbine is guaranteed to have an efficiency of at least 80 per cent. when delivering 6000 horsepower, at least 82 per cent. when delivering 5000 horsepower, and 80 per cent. when delivering 4000 horsepower.

Applying Patches to Boilers

By H. S. Jeffery

The method used in repairing a boiler plate depends upon the nature and location of the defect. Cracks may frequently be repaired by calking, while in other cases plugging has to be resorted to; or, if accompanied by a deformation of the plate, a patch may have to be applied.

When it is necessary to patch a boiler in a section adjacent to the girth seam, a horseshoe patch may be employed, the defective plate being cut out in such a manner that a number of the girth-seam rivets are removed, and when replaced hold part of the patch in place. At *A*, Fig. 1, is shown the usual method of cutting out the defective plate. This method is recommended over that shown at *B* as it does away with a seam in the longitudinal plane.

When the patch is applied in this manner and on the inside of the boiler, to the small course, the shell sheet should be scarfed, as shown at *A*, Fig. 2. Should the patch be applied to the large course, then it must be scarfed instead of the shell sheet, and the inside and outside courses forced apart sufficiently to insert the patch between them, the whole being fitted as shown in Fig. 3.

When the shell sheet is scarfed, the operation has to be done when cold, and consists of tapering the edge of the sheet

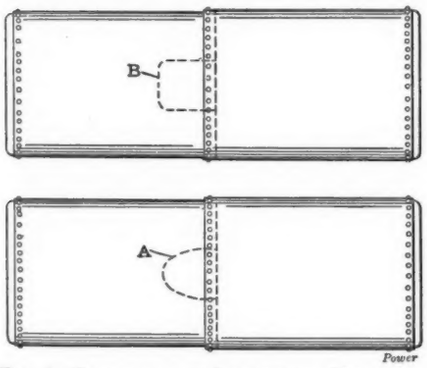


FIG. 1. CORRECT AND INCORRECT METHODS OF CUTTING

with a chisel, as at *A*, Fig. 4. This amounts to merely removing sufficient metal to prevent offsetting of the patch. On the other hand, when the patch is to be scarfed, the part concerned is heated and while hot is drawn out.

In order to insure a close fit the patch must be formed to whatever shape is required, and must be free from burs. The shell should also be free from all burs or foreign substances. The rivet holes may be either punched or drilled. If the work is performed in the boiler shop, then as many holes as possible, unless otherwise specified, may be punched by a power punch, and the balance punched by a hand punch or drilled. If the holes are to be drilled, it is advisable to employ guide lines, as the drill is liable to run to one side, and the guide line will indicate whether the drill is cutting the hole true.

Frequently it happens that the holes marked on one side of a plate can best be made from the other side, in which case the center-punch marks are transferred by the use of a tool, shown in Fig. 5, and called a "backmarker." The operation consists of placing the small hole *A* directly over the center-punch mark, and then marking through hole *B* on the other side.

PATCH ADJOINING THE HEAD

A patch on the shell sheet of a tubular boiler, having its smoke box formed by extending the shell sheet, as shown in Fig. 6, should be applied on the inside and scarfed to the calking edge *A*, thus permitting the rivets *B* to pass through the head, the patch and the shell sheet. Since the scarfed end of the patch is inserted between the head and the shell sheet, the latter will be forced downward at *C*. Before the rivets are driven the shell sheet should be fitted close to

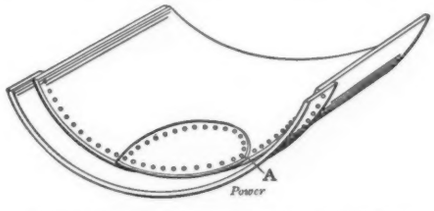


FIG. 2. PATCH WITH SHEET SCARFED

the patch, and the latter close to the head. The rivets should be driven from the outside of the boiler, both on account of accessibility and for the reason that the rivet hole on the fire side will be filled.

When a tubular boiler is constructed with the flange of the head inside the boiler the patch is applied inside the shell sheet, with the end of the patch scarfed and inserted between the head and shell.

PATCHING THE HEAD

Frequently a portion of the flange of the head of a boiler, tank or air reservoir cracks, or the metal so suffers from corrosion that a patch is necessary. When the defective part has been cut out, the patch should be shaped so as to be riveted partly to the head, as shown in Fig. 7. When a patch is thus applied, the ends of the patch are scarfed and inserted between the shell sheet and the head, and the latter at that point is slightly forced up. In every case the holes should be fair and the rivets well driven so as to completely fill the hole. The calking should be done with a round-nosed fuller and the lap should not be greater than one and one-half times the diameter of the rivet hole.

CRACKED FURNACE SHEETS

The firebox or furnace sheet of a locomotive, traction, or similar type of boiler frequently suffers from cracks at the stay-bolt holes, as shown in Fig. 8; these cracks often extending from one stay-bolt hole to another. How to best repair a fracture of this nature depends upon the length and the location of the crack. Frequently the crack can be closed by calking and gives no further trouble, but a long crack in any part of the firebox, however, cannot be calked over. This is also true of a short crack which is in contact with or very near the bed of the fire.

A small crack is often made steam tight by plugging, as shown in Fig. 9. The hole is usually drilled for a ½-inch tap, a solid plug inserted and then calked over. If more than one plug is needed, which would be the case with a long crack, the holes should be drilled so as to overlap. However, all the holes should not be drilled at one time. Assuming the crack to extend from stay bolt to stay bolt, the holes should be laid off with about ¾-inch centers. They should then be tapped, the plugs inserted and cut off to the desired length, usually about ⅛ inch beyond the sheet. Holes are then drilled between them so as to take in a portion of the adjacent plugs. These holes are tapped and plugs inserted and cut off as in the first instance. The entire row is

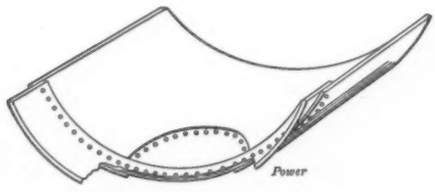


FIG. 3. PATCH INSERTED BETWEEN THE INSIDE AND OUTSIDE COURSES

then calked close to the sheet and forms a solid ridge, thus practically locking the plugs together. This operation is spoken of as "sewing up the crack." The work requires time and skill, and if satisfactory results are to be obtained, the plug

centers should be as uniform as practicable.

Bagging

The cracking of the furnace sheet is not the only cause for removing a part of the sheet. The plate may suffer from

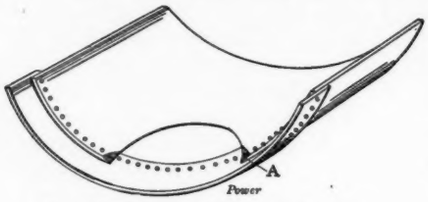


Fig. 4. Plate Properly Scarfed

bagging to the extent that a portion of it may have to be cut out and a patch applied. Bagging is usually caused by overheating of the plate, or by too great a spacing of the stay bolts. If the water space or water leg, as it is sometimes called, is allowed to clog up with foreign substances, such as mud and scale, then the sheet will become overheated and bagging will result. Where a sheet bulges between the staybolts due

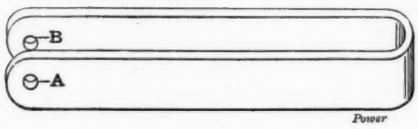


Fig. 5. Backmarker

to this cause, the sheet is said to be "mud-burnt," and, in the course of time, the stay bolts pull through the sheet. The spacing of the stay bolts too far apart, even though they are strong enough to carry the load, will result in bagging of the plate. The size of the stay bolts and their pitch must be in accord with the thickness of the plate that has to be supported.

Applying a Patch

Firebox patches should be riveted whenever possible, although they are frequenty applied with patch bolts. Applying a patch so as to utilize the stay bolts in lieu of rivets or patch bolts does not give satisfactory results owing to the difficulty of keeping the seam of the patch steam tight; the stay bolts being screwed into both the patch and the sheet and

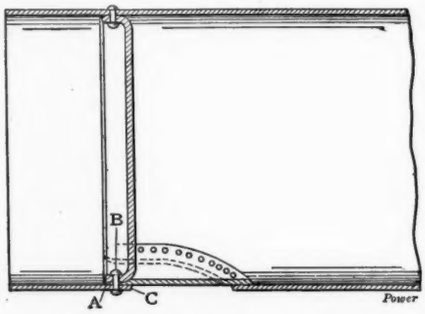


Fig. 6. Method Used When Smokebox is Continuous with Shell

hence not permitting the patch to be drawn tightly against the sheet. Even if this should be accomplished the pitch is excessive; that is, a 2-inch pitch is too great for ¾-inch patch bolts.

Whether riveted or held with bolts, a patch should be applied as in Fig. 10, which shows half a patch. The first operation is to cut out the defective part, and if the sheet is cracked it is well to cut out an inch or so beyond, for in many cases the fracture extends further into the sheet than can be seen from a surface inspection. The dotted line indicates how and where the sheet should be cut in regard to the stay-bolt holes, and

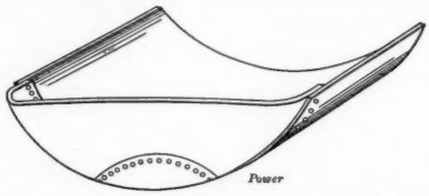


Fig. 7. Patch on Flange of the Head

to avoid sharp corners. The distance *A* is approximately 75 per cent. of the pitch of the stay bolts. The distance *B* must be sufficient to permit a cape chisel and a ripper to clear the stay bolts. The distance *C* is no set standard. Some boilermakers make this distance 1½ times the diameter of the patch bolt or rivet, though more often this figure is 1⅛.

The pitch of the patch bolt or rivet should be as uniform as practicable, the better practice being to lay off the holes, say for one quarter, and make the other

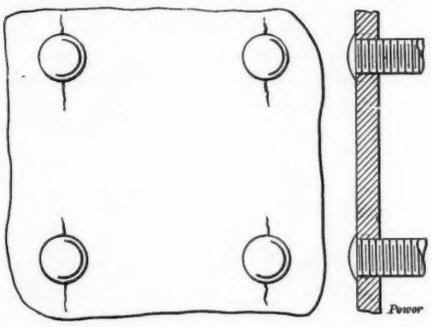


Fig. 8. Cracks around Stay Bolts

quarters accordingly. Some cut the holes in the furnace sheet first, and then with a sheet of heavy paper make a templet of the holes and drill the patch accordingly. A better practice, however, especially if patch bolts are to be used, is to drill a few holes in the furnace sheet to allow the patch to be bolted in place, and then drill the balance of the holes. Still another custom is to punch small holes in the patch, bolting the latter to the sheet as described, and drill the holes in the furnace sheet from the patch. The holes in the patch prevent the drill from running out, which is liable to occur if the patch is bolted in place with a few bolts, when drilling the holes in the patch and the furnace sheet in one operation. However, both of the last two methods insure fair holes, and are superior to the paper-templet method.

If the centers of the respective holes are in line and the holes in the patch are properly reamed out, no trouble should be experienced in keeping the alinement.

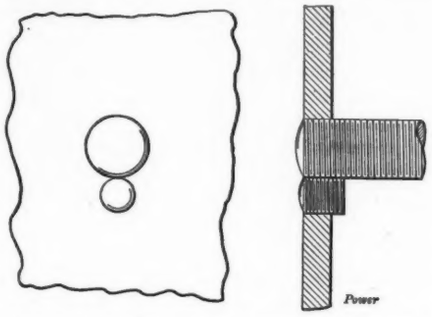


Fig. 9. Plug Inserted to Stop Crack

The countersinking of the holes in the patch is usually performed on a drill press, there being no objections to this practice if the patch is to be applied with rivets. But when patch bolts are used the above method of countersinking is not advisable, because of the difficulty in tapping the holes in the furnace sheet so the countersunk head of the patch bolt will fit the countersunk hole as intended.

The above difficulty is overcome by tapping the holes and then countersinking them while the patch is in place, a special tool with a guide being employed. Following the countersinking of the holes when the patch is bolted in position, the patch should be removed and all burs, cuttings, etc., removed, so as to leave

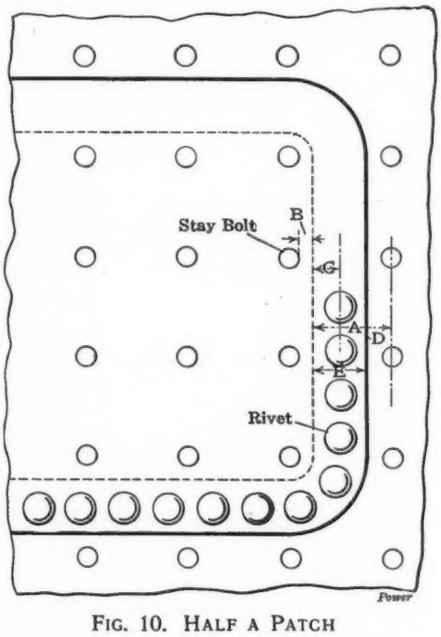


Fig. 10. Half a Patch

nothing in the form of foreign substances between the patch and furnace sheet, unless it be a strip of copper, which is sometimes used as a gasket.

The use of a strip of copper is advocated by some boilermakers and con-

demned by others. If used it should be thoroughly annealed and placed only at the lap, that is, its width should be equal only to the distance *E*. Fig. 10, the copper line not extending to the calking edge *D*.

Applying the Patch Bolts

It is not good practice to completely insert one patch bolt before applying the

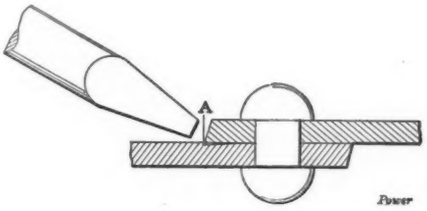


Fig. 11. Improper Calking Tool

next. Instead, they should all be partially screwed into place and then gradually tightened up and the square head twisted off. To facilitate twisting off the head without undue strain, the neck is nicked with a chisel. A calking tool, called a "fuller," is used to work the edges of the patch bolt to the patch. After the bolt has been fullered down, another calking tool, called a "frenchman," is used to cut away the edge of the bolt, but the cut in operation does not imply a recess, which is often made by inexperienced workmen.

The patch may or may not have been leveled before finally applying, and if not, it should be chipped and calked. If chipped in place, there is no reason why the furnace sheet should bear marks of the chisel, providing the calking is done with a fine finishing tool.

A Soft Patch

What is known as a soft patch is a patch applied over a hole or crack without cutting out the defective part, but it should never be used on a furnace sheet or where it will come in contact with intense heat. At the best it should be em-

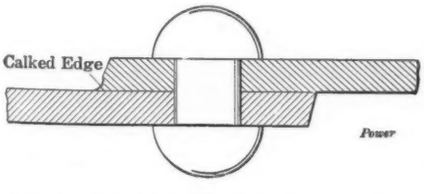


Fig. 12. Edge Upset with Round-nosed Fuller

ployed only as a means of temporary repair. To install a soft patch so as to require no calking, it should be a close fit, and the inside of the patch and the outside of the sheet should be coated with cement paste. Care should be taken to see that the surfaces are free from oil or grease, to insure the adhesion of the cement. When the patch is in place with the cement paste between the patch and sheet, the bolts should be tightened up to within about ⅛ inch of home. The joint, depending upon the class of cement paste, should not be finally finished at the outset, it being the better practice to allow the cement paste to set slightly and then screw up on the nuts. If possible, the patch should stand ten hours before the boiler is placed in service.

Leaky Firebox Seams

Many leaks in firebox seams are due to faulty construction, the calking very often being to blame. The shape of the calking tool is an important factor, likewise the position in which it is held. Usually the plate is beveled, not to exceed 30 degrees. A square-end calking tool, as shown in Fig. 11, should never be used as it will cause the sheet to be calked as at *A*, cutting into the plate and forming a ridge along the edge.

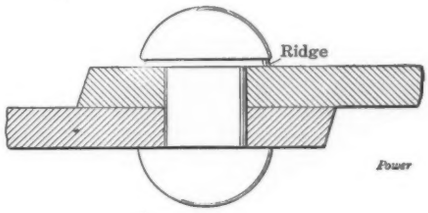


Fig. 13. Ridge Made by Calking Tool

The seam should be calked with a round-nosed fuller, thus upsetting the plate, as shown in Fig. 12. This tool causes the swelling to extend further into the plate, though care should be taken not to use too small a fuller—the size depends upon the thickness of the plate that is to be calked—otherwise it will act as a wedge and press the outer surface up.

The proper calking of a leaky seam, which is subsequent to the original calking, is a matter of utmost importance. If the leak is a small one, it can be stopped by recalking, that is, without chipping the sheet. The operation should be begun an inch or so beyond the leak and directed toward it with very light blows. Frequently, if the fuller is placed directly over the leak and a hard blow is struck, the plate at that point will be calked, but a new leak will appear on each side of the old one.

If the seam leaks to any extent it is the best practice to chip and then calk it, ⅛ inch usually being sufficient to remove the recess formed by the original calking. When chipping, care must be exercised not to cut into the sheet.

Leaky Rivets

Any rivet that leaks does not thoroughly fill the hole, and the calking of the rivet head is nothing more than fencing in the leak. The water and steam often find an outlet through the small spaces existing between the rivets and rivet holes. As the ratio of volume between steam and water is about 1700 to 1. at a temperature of 212 degrees, but very little water is needed to cause the impression that the leak is one of great proportion.

A button-head rivet whose edges have

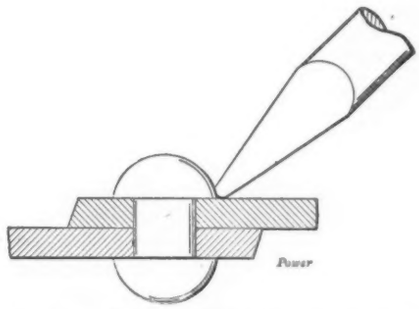


Fig. 14. Proper Method of Calking

not been brought down well to the sheet, as shown in Fig. 13, is the most difficult kind to calk, as the ridge becomes enlarged each time the operation is repeated. The rivet head should be calked as shown in Fig. 14, and while this method tends to pry up the rivet head, the effect is not as pronounced as in Fig. 13.

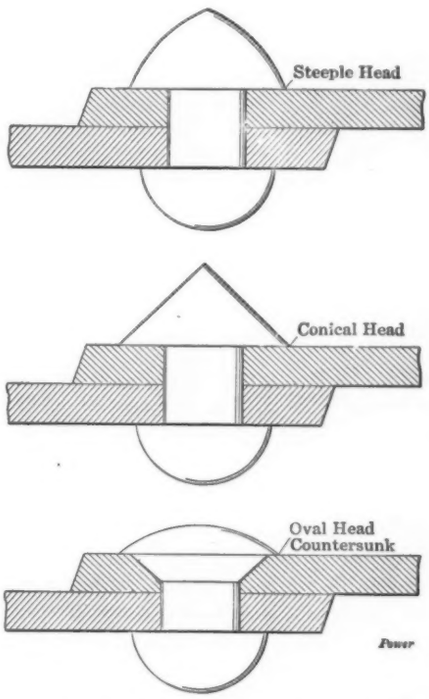


Fig. 15. Common Types of Rivet Head

The oval-head countersunk rivet, the steeple head and the concial head, the latter being a compromise between the button head and the steeple head, are more readily calked, for the reason that the calking tool can be readily applied to the edges. These types of rivet are shown in Fig. 15.

Central Station Steam Heating*

By A. D. Spencer

Description and operating cost details of a live-steam central-station heating system and an exhaust-steam central-station heating system in Detroit, Mich.

The Central Heating Company, of Detroit, Mich., operates two heating systems. System *A* is located in the business section. At the present time it operates with live steam. The plant is designed, however, for the ultimate installation of noncondensing turbo-generators. During the past winter the system served 400,000 square feet of radiating surface, and furnished steam for power and other purposes equivalent in amount to that required for about 100,000 square feet of radiation.

System *B* is located in a residential section, and uses the exhaust from three McIntosh & Seymour single-cylinder engines which are direct connected to generators and which are capable of developing 1300 kilowatts against the back pressure of the heating system. Service is furnished for about 300,000 square feet of radiation. Babcock & Wilcox boilers equipped with Jones stokers are used. The boiler capacity is 2465 horsepower. The station is equipped with 1000-kilowatt capacity of motor-generator sets and is connected to the 4600-volt transmission system of the Edison Illuminating Company. The generating equipment is operated whenever there is a demand for exhaust steam, some of the direct current generated being used in the district and the balance converted and delivered to the 4600-volt transmission system. When the district demand for electricity is greater than the equivalent demand for exhaust steam, the excess current is taken from the transmission lines and converted. When the demand for steam exceeds the capacity of the engines, live steam is used. In brief, the engines are operated as nearly as possible to furnish exhaust-steam requirements and no more. With the present heating load it is necessary to furnish some live steam during the winter months. On the other hand, it is generally necessary to use some electricity from the transmission lines during the lighting peaks.

The reasons for selecting a steam rather than a hot-water system are: A smaller investment is required; steam is capable of serving other requirements, such as for laundry work, cooking and power; steam can be adapted to hot-water installations by the use of transformers; and to hot-air systems by the use of indirect radiation; as hot-water service cannot be metered, charge for it must be made on a basis of radiation or on some other approximate plan.

TABLE 1.
OPERATING DATA.

1909.	System "A."	System "B."	Combined.
Pounds steam sold—heating	167,700,000	141,700,000	309,400,000
Pounds steam sold—power, etc.	41,300,000	5,250,000	46,550,000
Total steam sales	209,000,000	146,950,000	355,950,000
Electricity sold (kwh.)		4,523,200	4,523,200
Earnings per M. lb. steam	47.5	48.2	47.8
Earnings per M. cu.ft. space	2.10	3.90	3.0
Earnings per sq.ft. radiation	21.4	23.7	22.6
Number of customers	274	396	670
Cubic feet space	37,200,000	17,300,000	54,500,000
*Square feet radiation	372,400	287,800	660,200
Ratio radiation to space	1–100	1–60	1–83
Cost of coal per ton	2.50	2.55	2.53
Lb. coal per M. lb. steam sold	220	220	220
Lb. steam sold per lb. coal	4.6	4.6	4.6
Lb. steam sold per M. cu.ft. space	4,500	8,200	6,350
Lb. steam sold per sq.ft. radiation	450	490	470
Heating efficiency, per cent	37	37	37
Steam sold in per cent. of generated			55.2
Steam used in auxiliaries in per cent. of generated			19.2
Line losses in per cent. of generated			9.8
Total sold or accounted for in per cent. of generated			84.2
Unaccounted for in per cent. of generated			15.8
			100.0

*In these figures no allowance is made for hot water heaters, of which there are about 200 on System "B" and 100 on System "A."

SYSTEM *A*

The plant of system *A* is equipped with eight 500-horsepower Stirling boilers Jones stokers, forced draft, feed-water heaters and ash-handling apparatus. The building contains coal-storage bins of 1500 tons capacity and is equipped with cranes for handling coal from the alley in five-ton hopper wagons.

DISTRIBUTION LINE

In a live-steam system, consideration of back pressure is not present; hence, the allowable line pressure is limited only by distributing conditions. The higher the pressure, the smaller the size of the distributing mains. On the other hand, increased pressure means increased trap troubles and other troubles in customers' installations. Fifteen pounds, average, in the heating mains was chosen as a sufficient pressure to handle some old 15-pound installations and to do cooking and miscellaneous service. The customers having lower-pressure systems were required to install regulating valves to reduce the pressure. The distributing pressure has been increased from year to year, as increase in load made it necessary to increase either the pressure or the size of the mains.

To furnish power to operate steam pumps, and for other purposes, a system of 110-pounds pressure was installed in the heart of the business district. Incidentally, this system was designed for use as a feeder to the low-pressure system. The 110-pound service has proved extremely unsatisfactory, principally for lack of a steam meter.

Tunnel construction is used in the district where high-pressure steam is furnished. This opens the high-pressure mains to inspection and repair. The tunnel system is 3600 feet long and is at an average depth of 30 feet below grade. The standard tunnel is horseshoe shaped and 5 feet high. The construction is brick and hydraulic cement. Manholes are located about every 300 feet. Services are taken off at service manholes which are located wherever needed. Drainage flows to a sump at the plant and is siphoned out. An exhaust fan at the plant maintains a circulation of air and prevents accumulation of moisture. The normal temperature in the tunnel is about 150 degrees Fahrenheit, which is reduced by local ventilation when work has to be done in any section.

The 110-pound system consists of 3300 feet of 6-inch and 8-inch mains covered with magnesia and some hair felt; 1100

TABLE 2.

Cost in cents per 1000 pounds of steam sold in 1909.

	Plant A.	Plant B.	Average.
Production	32.7	38.0	34.9
Distribution	2.9	3.1	3.0
Sales and collections	0.4	0.4	0.4
General	2.5	3.6	3.0
Injuries and damages	0.1	0.1	0.1
Insurance and taxes	4.2	4.7	4.4
Electricity sold (credit)		(15.4)	(6.4)
Total of above items	42.7	34.5	39.5
Depreciation	6.7	8.9	7.6
Interest	11.6	16.1	13.1
Total cost	61.0	59.5	60.2
Profit to make total returns on investment 8 per cent	6.2	8.2	7.0
Total	67.2	67.7	67.2
Earnings	(47.5)	(48.2)	(47.8)

*Abstract of paper read at the meeting of the National District Heating Association, at Toledo, Ohio, June 1, 2 and 3.

feet of 1-inch to 3½-inch service connections and a 10-inch tie line 4100 feet long connecting plant *A* to system *B*. This is to assist in maintaining pressure in the nearest part of the *B* system and serves also to divide the load between the two plants in the manner which at the time is most economical.

The low-pressure system in the tunnel consists of 3500 feet of 6-inch to 12-inch pipe mains covered with magnesia; also 1800 feet of 1-inch to 4-inch services. In addition to the tunnel construction and the tie line, there are 18,700 feet of underground mains of various sizes from 2½ inches to 8 inches and 91,300 feet of services from 1 inch to 6 inches in size. The construction is full-weight wrought-iron pipe insulated with asbestos paper and wooden log. Expansion joints of the copper-diaphragm type are provided every

TABLE 3.

Cost (in cents) per square foot radiation.

1909.	A.	B.	Average.
Production	14.7	18.7	16.7
Distribution	1.3	1.5	1.4
Sales and collections	1.2	1.2	1.2
General	1.1	1.8	1.4
Injuries and damages	0.0	0.0	0.0
Insurance and taxes	1.9	2.3	2.1
Electricity sold (credit)		(7.6)	(3.0)
Total of above items	19.2	17.0	18.5
Depreciation	3.0	4.4	3.6
Interest	5.2	7.9	6.3
Total cost	27.4	29.4	28.4
Profit	2.8	4.0	3.3
Total	30.2	33.4	31.7
Earnings	(21.4)	(23.7)	(22.4)

65 feet and pipes are anchored at these points. Manholes are provided in each block and mains are graded to cause line condensation to flow to these manholes, whence it is trapped to the sewers. All underground construction is provided with tiled drain and is laid in 2 inches of crushed stone and covered with tar paper. Service connections are taken off from the top of the pipe at anchored points. Steam is furnished on all lines from October 1 to June 1. Lines carrying power are kept alive throughout the summer.

System *B*

The distribution arrangements of system *B* are practically the same as those of system *A*, except that there is no tunnel construction and, as the distribution pressure is but five pounds, the mains are considerably larger.

There are in this system 29,900 feet of 2½-inch to 18-inch low-pressure mains and 26,900 feet of 1½-inch to 5-inch services; also one 4-inch, 80-pound line 800 feet long laid in concrete, furnishing heat and power to a hospital.

The weak points of this system are in the customer's installations. Regulating valves fail to operate, causing excessive high or undesirably low pressure. Traps fail to open, thus flooding the system, or fail to close, thus allowing steam to be lost to the sewer. Automatic air valves fail to open, causing an accumulation of air in the system. Meters operate unsatisfactorily under the extremely severe conditions of heat, moisture, dirt and steam from defective traps.

Efforts are made to minimize these troubles by inspecting installations before service is turned on and advising customers to use proper equipment and methods of installation, and by refusing to connect service when, in the judgment of the company, the installation is not in condition to give satisfactory results to the customer or to the company. The company carries repair parts for certain lines of valves and traps, and makes all necessary repairs, charging to customers the cost of the material only. No other makes of valves and traps will be approved unless customers employ an engineer competent to make repairs. Continual study has been made of the meters and considerable improvement in metering has been effected.

Meters are read and inspected every six days, and traps, valves or meters out of order are quickly brought to the attention of the company and repaired. In case a meter is stopped it is necessary to estimate the consumption for a few days only and not for the entire month.

Twenty-four-hour emergency service is maintained and all ordinary installation troubles are remedied by the emergency men.

Steam Losses

Table 1 gives steam sold, steam lost, etc. The steam used in auxiliaries is practically all used to heat the boiler-feed water as the auxiliaries exhaust into the heaters. At plant *B* the heater is connected to the exhaust heating line, and the feed water is brought to a temperature of about 220 degrees.

Steam loss is calculated by the use of a radiation factor determined by actual metering of line losses. For a number of years meters were maintained on line bleeders, and actual line condensation per square foot of line radiation was determined. The condensation per hour per square foot of radiation was found to be 0.086 pound on 20-pound lines and 0.051 pound on 5-pound lines. These factors are for log construction. No data have been obtained on concrete construction. Radiation losses in the tunnel have been calculated from handbook data. Some part of the unaccounted-for steam may be due to imperfect metering, to

TABLE 4.

Present Rates.

Monthly bills for steam consumption aggregating—	Gross.	Net.
0 lb. to 12,500 lb	58	52.2
12,500 lb. to 25,000 lb	57	51.3
25,000 lb. to 37,500 lb	56	50.4
37,500 lb. to 50,000 lb	55	49.5
50,000 lb. to 75,000 lb	54	48.6
75,000 lb. to 100,000 lb	53	47.7
100,000 lb. to 200,000 lb	52	46.8
200,000 lb. to 300,000 lb	51	45.9
300,000 lb. to 400,000 lb	50	45.0
400,000 lb. to 500,000 lb	49	44.1
500,000 lb. to 600,000 lb	48	43.2
600,000 lb. to 800,000 lb	47	42.3
800,000 lb. to 1,000,000 lb	46	41.4
Above 1,000,000 lb	45	40.5

Minimum monthly charge, 5,000 lb.

steam losses by leakage, and to errors in estimating evaporation. It is possible that the progressive deterioration of the wooden-log insulation has increased the radiation losses.

Costs

Operating costs are set forth in Table 1. Table 2 gives the cost per thousand pounds of steam sold. In Table 3 the cost per square foot of radiation is given, and Table 4 gives the existing scale of charges per thousand pounds of steam consumed. In establishing these rates, the idea was to make the cost to the customer about the same as his own cost had been, as it was believed that, costs being equal, the advantages of the service would bring the business. Data on 40 residences showed that the average cost of heating by coal in Detroit was 4.60 dollars per thousand cubic feet of space, with coal selling at 6.75 dollars per ton. Comparing this figure with the actual earnings for system *B*, it is seen that the actual earnings are 15 per cent. under this. At the present price of hard coal, 7.50 dollars, the customer's cost would be 5.10 dollars per thousand cubic feet.

The following rules were recently found hung on the wall of a large power plant in Richmond, Va., which, if observed, would doubtless eliminate a great many accidents in the boiler room:

Do not empty the boiler while the brickwork is hot.

Do not feed cold water into a hot boiler.

Do not allow filth to accumulate around the boilers or boiler room.

Do not leave your shovel or any tool out of its appointed place when not in use.

Be alert and ready-minded about the boiler and furnaces.

Do not read newspapers while on duty.

Do not fail to see that there is plenty of water in the boiler in the morning.

Keep the water level at the same hight in the boiler during the running period.

Do not allow anyone to talk to you when firing.

Do not allow water to remain on the floor about the boilers.

Do not fire too quickly.

Do not fail to blow off the boiler once or twice a day, depending on the condition of the water.

Do not fail to close the blowoff cock when blowing down, when the water in the boiler has lowered 1½ inches.

Daily see that all safety valves move freely and are tight.

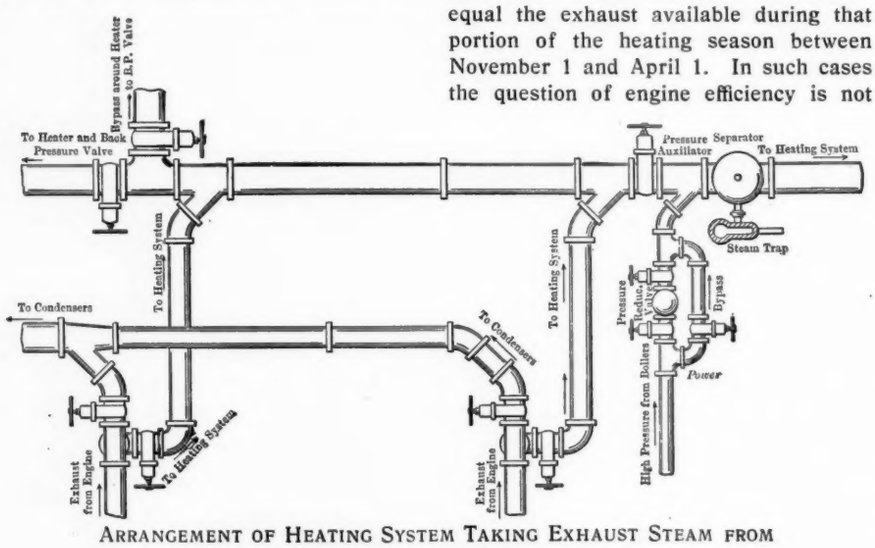
Factors Affecting Cost of Heating*

By Charles R. Bishop

For the successful operation of a heating plant or a combination heating and electric generating plant, all the factors entering into the generation and utilization of the steam and the maintenance of the system should be carefully noted. For this purpose complete daily records are an absolute necessity.

The first factor requiring consideration in any heating plant is the thermal value of the fuel burned, whether it be coal, wood, oil or natural gas. Many companies are now purchasing coal upon a basis of analysis, but where this is not practicable, it is advisable to thoroughly test the different grades under actual operating conditions to determine the number of pounds of steam which can be produced not only per pound of fuel burned, but more particularly, per dollar expended.

With a good quality of fuel and water and a generating plant containing economical boilers of the proper size, attention should be given to the means of utilizing the greatest percentage of heat units in the fuel to be burned. In this connection, it is considered good practice to equip plants of 1000 or more horsepower with mechanical stokers adapted to the kind of fuel used. Such an installation not only increases the efficiency but decreases the firing cost per ton. In hand-fired plants, one man can attend to the coal, water and ashes for 200 horsepower, while with mechanical stokers and overhead bunkers he can easily handle from 2000 to 3000 horsepower. It is also advisable to install coal- and ash-handling apparatus for cutting down the labor costs.



ARRANGEMENT OF HEATING SYSTEM TAKING EXHAUST STEAM FROM SEVERAL ENGINE UNITS

Proper feed-water heating and clean boilers will also result in a marked fuel economy.

After considering the factors which enter into the generation of steam, the next question to be taken up is its utilization. If the plant in mind is a combination electric-generating and steam-heating plant, it will first be assumed that it intends to have or does have a market sufficient to utilize the full amount of exhaust steam at such times as it is developing the greatest electrical load, and that its steam demand will at least equal the exhaust available during that portion of the heating season between November 1 and April 1. In such cases the question of engine efficiency is not as serious as engine capacity, up to a point of approximately 65 per cent. of engine capacity, although this figure varies with different companies, due to the proportion of summer to winter maximum electric load. Some companies desire to install a sufficient number of high-efficiency engines or turbines to permit of summer operation under economic generating costs, while others believe the installation of exhaust turbines for summer operation is advisable.

It is a fact that engines act, in a sense, as reducing valves between the boilers and the heating system, but while performing such functions produce mechanical energy without appreciably lessening the heat energy; therefore, were there a heat demand in excess of the amount of steam exhausted by a high-efficiency engine, the deficiency in steam would have to be made up direct from the boilers, and thus the quantity of steam generated in the boilers would not be less than if all had passed through the engines in the production of a given quantity of engine horsepower or its equivalent in kilowatt-hours. By this statement, I do not mean to convey the idea that it is advisable to install or continue to operate engines of very low efficiency; I desire to more particularly call to mind that it is not necessary to successful operation and low kilowatt cost in combination plants to install engines of the highest efficiency.

Where the station contains several engines, an additional exhaust main should be installed, so that any engine can be run condensing or exhaust to the atmosphere, while other engines are exhausting into the heating mains. Such an arrangement of piping is shown in the accompanying illustration.

Transmission losses in a heating system are affected by the following factors: Efficiency of insulation; outside water coming into contact with the distributing mains; air currents coming in contact with the distributing mains; and actual leakage of steam. Frequent inspection of the distributing mains should be made and street traps should be kept in perfect working condition.

Normal transmission losses are practically constant in properly installed mains and should never exceed 0.045 pound per hour per square foot. The percentage of loss to output depends upon the load factor and should not exceed 15 per cent. at times of minimum demand nor 1½ per cent. during periods of maximum demand with an average of about 4 per cent. during the heating season, which lasts about five months.

The rate charged the consumer, while taking into consideration the total cost to the company with profit added, must still be reasonable and fair and is naturally dependable upon local conditions. Sliding-scale rates are usually preferable, as they automatically fix a lower average price per 1000 pounds to the large consumer than to the small consumer.

The degree of success of a heating and power plant depends largely upon the manager's knowledge of all the conditions existing at the plant, in the distributing system and at the consumers' premises. Such information can be had and the economical operation of the plant maintained only by a system of complete daily records in such form as to be readily comparable for reference at all times.

*Abstract of a paper delivered before the National District Heating Association, at Toledo, Ohio, June 1-3, 1910.

The Operating Engineer's Problem

By Hubert E. Collins

The following thoughts were offered by Mr. Collins at a meeting of the Modern Science Club held recently to consider the proposed organization of an Institute of Operative Engineers. He points out the entirely inadequate remuneration received by operating engineers in comparison with those paid in other vocations, and how the proposed organization may be expected to improve their condition.

The editorial in POWER of February 8, 1910, issue, entitled "Can You Beat It," really brings up the most vital problem relating to the material welfare of the operating engineers' vocation. The correct solution of this problem is the aim and object of all engineering organizations and individuals who would better their own conditions and that of their fellow workers.

The fundamental fact brought out in this editorial was that engineers work longer hours and receive much less pay for services than many kindred workers who are tradesmen only. The case cited was not an isolated one, but is generally true.

An all-round engineer, for instance, is a trained machinist, for it is a part of his training to become familiar with this work. In this day of advanced operating engineering the man in charge has also been trained in other vocations.

To realize the comparative remuneration received by an operating engineer and a pipefitter, we have but to remember the training of the two.

The operating engineer first began as a machinist. While an apprentice in the machine shop he was taught pipe fitting better than 99 out of 100 pipefitters who learn that one vocation only. In fact, if this apprentice boy were kept at pipe fitting he would feel disgraced or he would never make a good mechanic. Now this training as pipefitter is only a small part of that of a machinist, and the machinist trade is only a small part of the equipment an operating engineer needs in an uptodate plant. Yet the fact remains that operating engineers as a class are paid less and work longer hours per day than steamfitters, plumbers, carpenters and masons. Many operating engineers could occupy a position in any one of these trades with credit, but how many members of any one of these trades could take the operating engineers' job? This is not mentioned to belittle the other trades. It is only to emphasize the fact which does exist. The operating engineer does not get the compensation due one who fulfils the requirements of a good man in this vocation. Many work on dissatisfied, knowing that this condition grows more acute as the years go by. Others become disgusted and leave the field for more profitable employment in other trades. Some who realize the conditions do what they can to aid themselves and others to better conditions in the profession.

The solution is to organize and make better engineers first; then better pay will come naturally. To organize first for better pay irrespective of qualification is wrong to both employer and employee if we are just. Before the organization of operating engineers is brought about for the permanent betterment of the vocation we must consider the elements that are to make up this body:

1. The theoretical man who enters the practical field better equipped for ultimate qualification in the practical field.

2. The practical man who requires enough training in the theoretical field to grasp all the principles of why and wherefore.

3. The employer.

The first two elements of this organization must meet on a common field to acquire that which they do not have to convince the third element that he is securing just what he wants in the way of all-round engineering ability.

The fraternal engineering associations have been trying for years to solve this problem and have fallen short. It has required more concerted effort than they have been able to maintain.

The Institute of Operating Engineers is the best effort so far conceived to bring about ideal conditions by which operating engineering can be placed on a professional plane and be recognized as such by the employers and general public.

The editorials in POWER, March 20 and May 24, as well as the editorial in the June *Practical Engineer*, very clearly set forth the plan and give an idea of the scope of this latest phase of engineering development. One thing we all see is that fundamentally it is the greatest move ever made for the good of the operating engineer. Another thing we see is that the more we think it over the greater the whole plan looks.

Those who are taking up the development of this plan are finding the work worthy of their best efforts and a truly herculean task, the accomplishment of which will be an honor to all concerned.

To the prospective members the requirements will seem hard, but if the plan of the Institute is carried out as it will be there will begin to crop up in a few years one of the best bodies of organized operating engineers in the world. The employer will recognize nembers of the Institute as the best operating engineers to be secured and will have them and no other to serve him in this department.

The idea is creating untold interest already. Many schools and colleges stand ready to establish classes on these lines as soon as the work is laid out for them, and in school centers it is a foregone conclusion that this will be an established Institute throughout the country as soon as it can be given them.

The Institute must provide for the member in isolated localities also where he will need to work his way by correspondence and home-study methods in connection with his practical training. But the plan is practical in all cases. It may interest many to know that J. C. Jurgensen, the proposer of the Institute, graduated a class from Columbia University this spring from the first year's work, on the lines laid down in the first year's work of the Institute of Operating Engineers. This class comprised twenty-eight members, one of whom was a college graduate, two firemen and twenty-five engineers. Every member was working and attending this class. Every one stuck it out and passed a creditable examination, securing the coveted first year's certificate. This shows that this work is far from being visionary. The Institute of Operating Engineers is started and will succeed. It will be the solution of the operating engineers' problem.

Blast-furnace creosote oil is produced in vast quantities in Scotland, and the question of application has received a considerable amount of attention. To a certain extent it has been used for creosoting timber, liquid fuel, and for illuminating purposes. It furnishes from 20 to 35 per cent. of phenoloid substances, soluble in caustic soda, as against 4 to 7 per cent. in London coal-tar creosote, and about 16 per cent. for Midland or county make. One ton of creosote is capable of yielding 13,300 cubic feet of gas, with an illuminating power of 14 candles, or 29,300 cubic feet at 8½ candlepower. Information from *The Consular and Trade Reports* indicates that the annual production of creosote oil in the United Kingdom is approximately 50,000,000 gallons.

Hydraulic Rams and Their Capacity

By N. A. Carle

For certain kinds of service the hydraulic ram is a very efficient piece of apparatus, and within the past few years they have been built in comparatively large sizes. To obtain the best efficiencies it is necessary that the proper factors be taken into consideration and to facilitate such operations the accompanying set of charts has been prepared.

A hydraulic ram is an automatic pump by which the fall or force of running water is utilized to pump a part of that water to a higher elevation.

The common idea of a hydraulic ram is that it is an extremely inefficient piece of apparatus suitable only for raising a small quantity of water for domestic purposes with a minimum amount of attention, practically no operating expense, and a low first cost of installation.

Hydraulic rams have been built for the past hundred years but it is only within recent years that they have been constructed in larger sizes and their efficiency materially improved. At the present time manufacturers are willing to contract for rams capable of delivering as high as 150,000 gallons of water per day of 24 hours, and with these units combined in batteries there is practically no limit to the size of the proposed installation, provided sufficient water is available.

These rams are guaranteed to operate on heads of from 2 to 50 feet at efficiencies ranging from 60 to 90 per cent., these efficiencies varying with the ratio of power head to discharge head. The usual power head for a ram is from 8 to 12 feet, because heads in excess of this are usually wanted for power purposes. Where all the power is required for pumping water, the modern hydraulic ram is more efficient for heads less than 50 feet than a waterwheel direct connected to a centrifugal pump or any arrangement of electrically operated pumps.

A ram will pump water to a hight of 45 feet for every foot of power head, but the reduction in the volume of the water pumped at this ratio is so great that it is not a practical operation. The ratio of 30 to 1 with a maximum lift of 500 feet is taken as the commercial limit of the ram.

There has been considerable discussion as to what factors govern the efficiency of a ram and the correct formula to be used in computing the efficiency. Rankine proposes the following:

$$E = \frac{q\,h}{(Q - q)\,H}$$

where,

E = Efficiency,
q = Quantity of water delivered per second,
Q = Quantity of power water used per second,
h = Difference in elevation between the water-supply level and the water level in the delivery tank,
H = Difference in elevation between the level of the water supply and that of the escape valve on the ram.

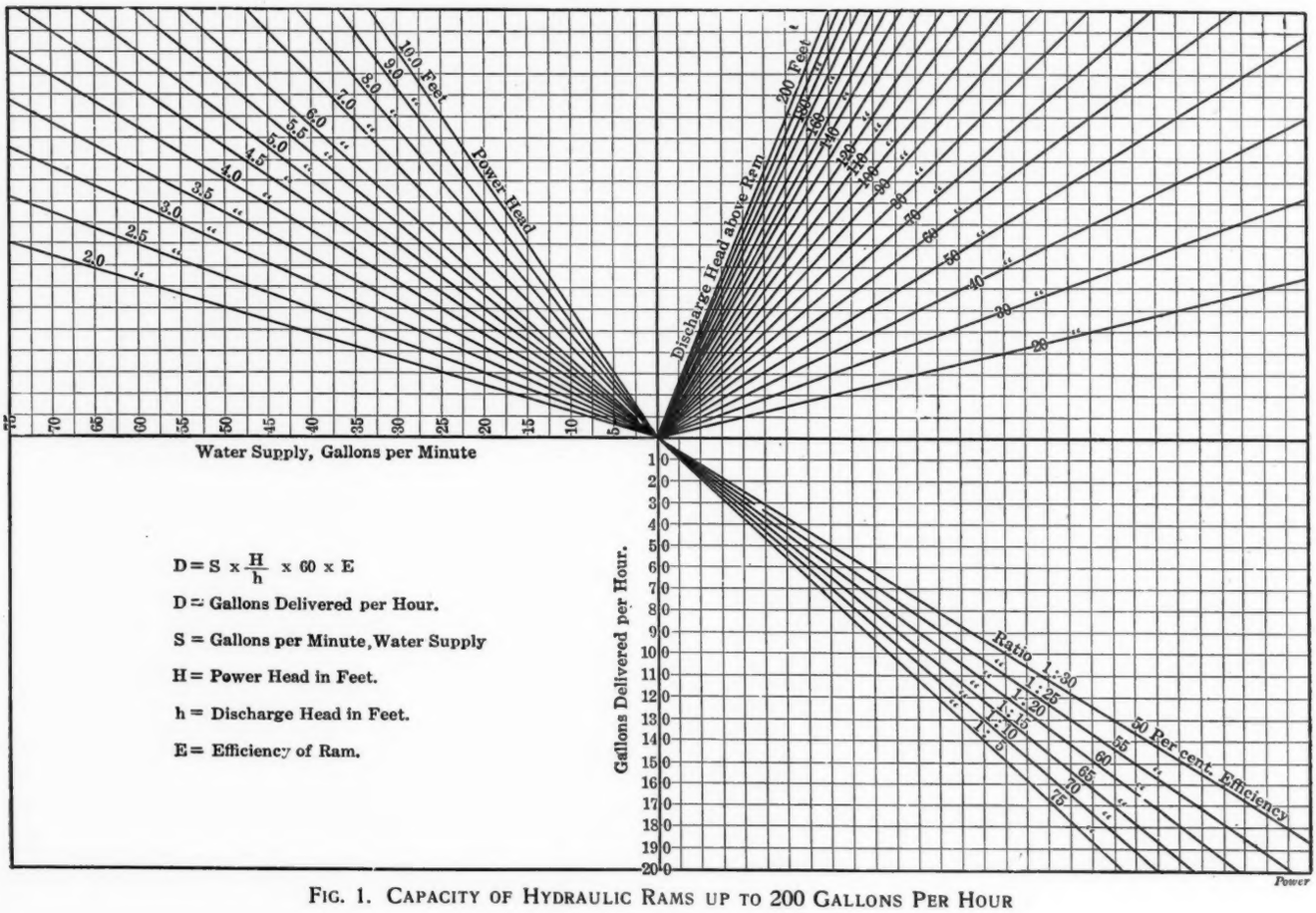


FIG. 1. CAPACITY OF HYDRAULIC RAMS UP TO 200 GALLONS PER HOUR

D'Aubisson proposes the following formula, the factors being represented by the same symbols as in the above:

$$E = \frac{q\,(H + h)}{Q\,H}$$

An analysis of these will show that Rankine's formula deals with the operation of elevating water from the supply reservoir to the delivery tank, while D'Aubisson's formula considers the ram as a machine receiving energy in the form of water under pressure and delivering some part of it at a higher pressure.

It is immaterial which formula is used to obtain the value of E, provided the formula for the capacity of the ram which uses this value of E is derived along the same lines.

The capacity of a ram is expressed by the following formula:

$$D = S \times \frac{H}{h} \times 60 \times E$$

where,

D = Quantity of water delivered in gallons per hour.

S = Quantity of water used in gallons per minute.

H = Difference in elevation between the water-supply level and the escape valve on the ram, commonly called the power head.

h = Difference in elevation between the escape valve on the ram and the water level in the delivery tank, commonly called the pumping head.

E = Efficiency of the ram calculated by D'Aubisson's formula, the values of which vary with the ratio of the power head to the pumping head as follows:

E = 75 per cent. when ratio = 1:5
70 per cent. when ratio = 1:10
65 per cent. when ratio = 1:15
60 per cent. when ratio = 1:20
55 per cent. when ratio = 1:25
50 per cent. when ratio = 1:30

The accompanying charts are intended to show graphically the quantity of water per hour and the hights to which it can be delivered by different amounts of power water used per minute under various heads. Chart No. 1 covers quantities of water supply up to 75 gallons per minute under heads of from 2 to 10 feet, delivering water up to 200 gallons per hour under heads of from 20 to 200 feet. Chart No. 2 can be used with water supplies up to 750 gallons per minute under heads of from 4 to 20 feet, delivering water in quantities up to 2500 gallons per hour under heads of from 40 to 400 feet. An example may serve to make this clear.

Example—With a power water supply of 30 gallons per minute and a 10-foot fall from the supply reservoir to the ram, how much water per hour can be raised to an elevation of 150 feet above the ram?

Solution—The ratio of power head to pumping head is 1:15; hence the efficiency of the ram will be 65 per cent. Using Chart No. 1, start with 30 gallons per minute, read up to 10-foot power head, then across to 150-foot discharge head, and down to 65 per cent. efficiency; following across to the left, the gallons of water delivered per hour will be found to equal 78.

The length of the supply pipe is very important as it governs the admission of air which is necessary for the efficient operation of the ram. Some manufacturers claim that the length of this pipe varies with the ratio of the power head to the pumping head. Successful installations show that the best results are obtained for small rams when the length of the supply pipe is four times the vertical fall, corresponding to an angle of 14.5 degrees with the horizontal, and five times the vertical fall, corresponding to an angle of 11.5 degrees with the horizontal, for larger rams.

The supply pipe should be of the same diameter throughout its length and without bends. The delivery pipe may be of any size suitable for carrying the water.

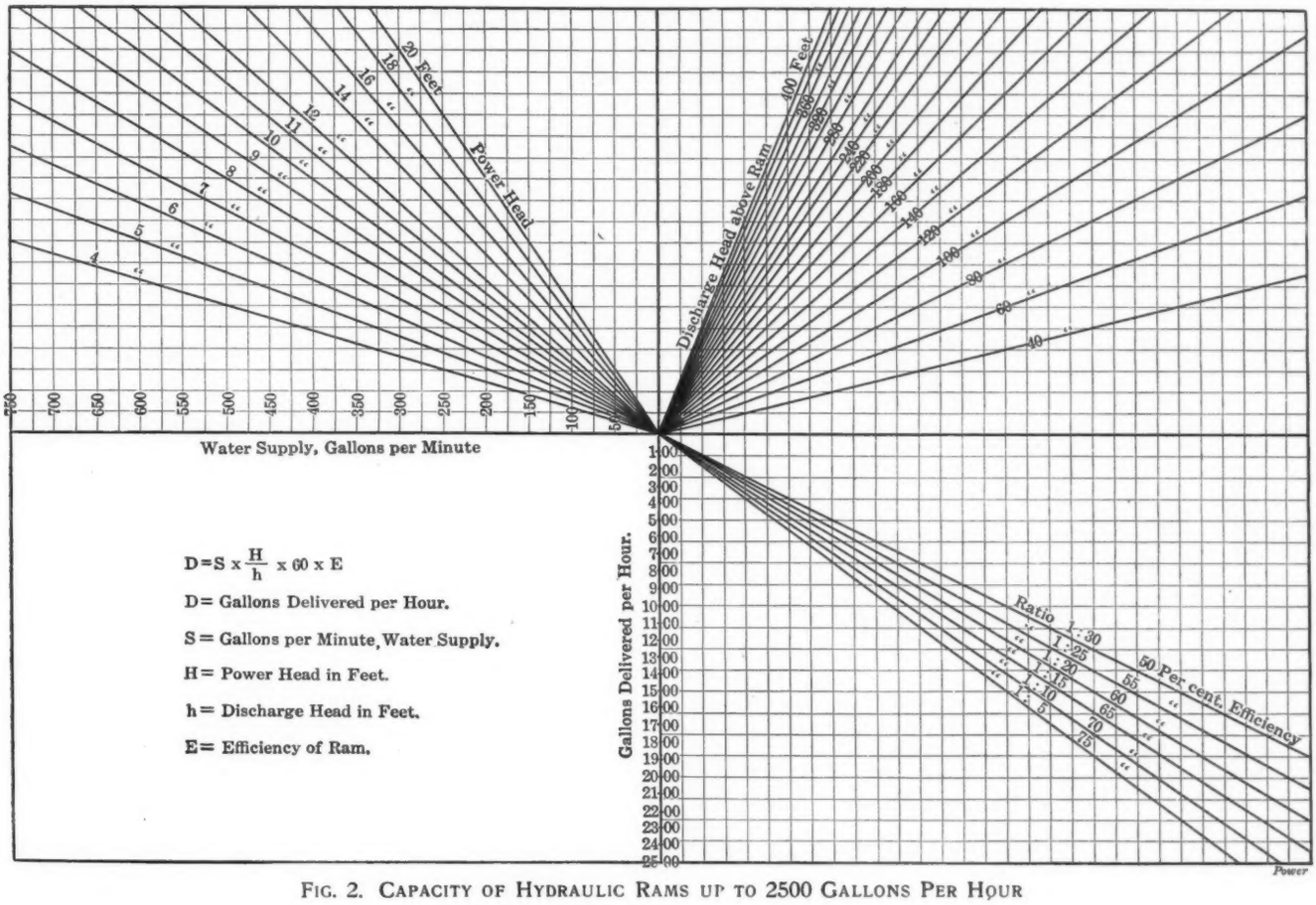


FIG. 2. CAPACITY OF HYDRAULIC RAMS UP TO 2500 GALLONS PER HOUR

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment.

The Electric Elevator

By William Baxter, Jr.

Elektron System with Mechanical Control

All the electric elevators described in previous articles belong to the type known as single-speed machines, which, after being brought up to speed run at substantially that speed constantly. Differences in load will make a slight difference in velocity but it is so slight as to be hardly noticeable. Electric elevators are also made to run at different speeds, and when so made are designated as "two-speed" or "three-speed," or as "variable speed" if they are arranged so as to run at any of several speeds, say, eight or ten. At the present time most of the elevators that are built for car speeds of 200 feet per minute or more are of the variable-speed type. The variations in speed are obtained in two ways, and both arrangements are very simple so far as the construction is concerned, but the principles involved in one of these methods are rather complicated, and I will not undertake to explain this method until machines in which it is used have been described.

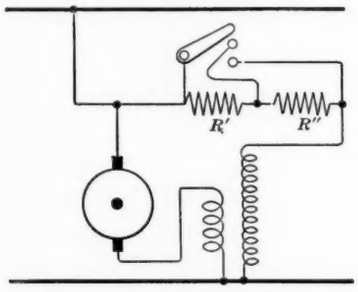


Fig. 112.

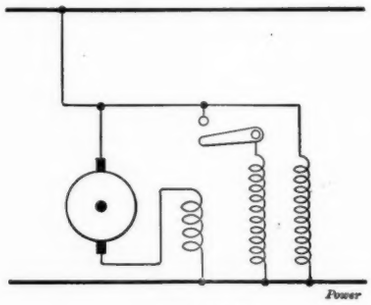


Fig. 113.

Methods for Varying Speed by Varying Field Strength

The other type operates on a principle that can be easily explained, and this one is considered here. The principle consists in varying the motor speed by changing the strength of its field excitation, and this change is accomplished in either of two ways, which are illustrated in the simple diagrams, Figs. 112 and 113. In Fig. 112, it will be noticed that there are two resistors R' and R'' connected in series with the shunt field winding. If these are short-circuited by means of the switch, the current that will flow through the shunt field coils will be stronger, of course, than if the resistors are in the circuit, and the strength of the field magnet will be greater than if the current is weakened by leaving the resistors in circuit; as a result, the speed of the motor will be low. Therefore, when the resistors are both short-circuited, the motor will run at its lowest speed. If the short-circuit around the resistor R'' is opened, so that the field current has to pass through this resistance, its strength will be reduced and, consequently, the field strength will be reduced and the motor speed will be increased. Opening the switch, entirely, as represented, so that both resistors are cut into the circuit, will still further increase the motor speed. This arrangement gives a three-speed system, and if only one resistor were used, the motor could be run at either of two speeds; with nine resistors and a controller made so as to cut these into the circuit one at a time, ten different speeds can be obtained, and then we will have what is called a "variable-speed" machine.

The speed variation that can be obtained in this way is not very great in practice, although it might seem that it could be made anything desired. To increase the speed range, the size of the motor must be increased, and while a range of two or three to one can be obtained without greatly increasing the size of the motor, a range of six to one would require a motor so large as to increase the cost of the elevator machine considerably.

Fig. 113 shows a way of varying the field strength that is just as effective as that of Fig. 112, but it is not so generally used. In this arrangement, there are two shunt field windings, one of which is the regular winding and the other an extra winding to be cut in or out for changing the speed. When the regular winding is in circuit alone, the motor speed will be greater than with both field windings connected, because the second winding strengthens the field and reduces the speed; if a third winding is added and current is passed through it, the motor speed will be still further reduced. One advantage of this arrangement is that as there are two or more field windings, there is very little danger of the field excitation suddenly dropping down to

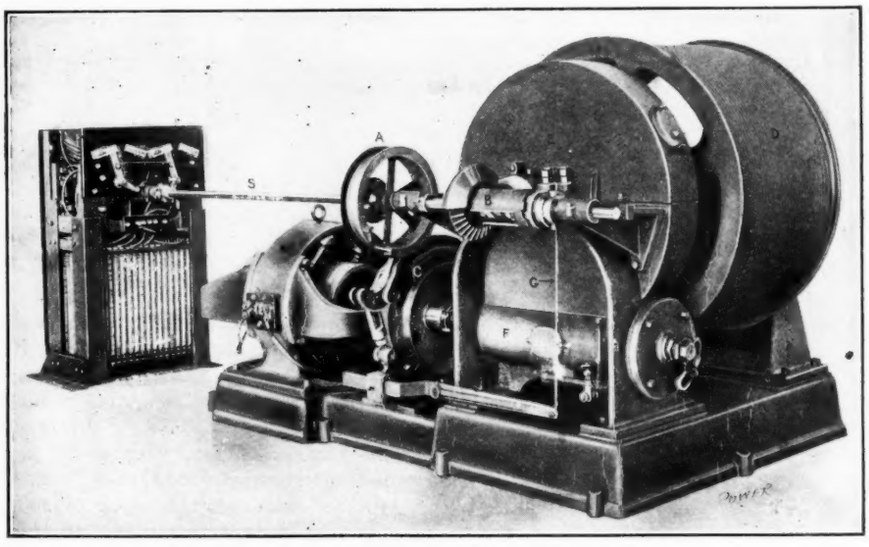


Fig. 114. Elektron Elevator with Mechanical Control and Automatic Acceleration

nothing, as it would if there were only one winding and the circuit of that one should be accidentally opened. With one field winding permanently connected, as in single-speed machines, there is less danger of its becoming open-circuited

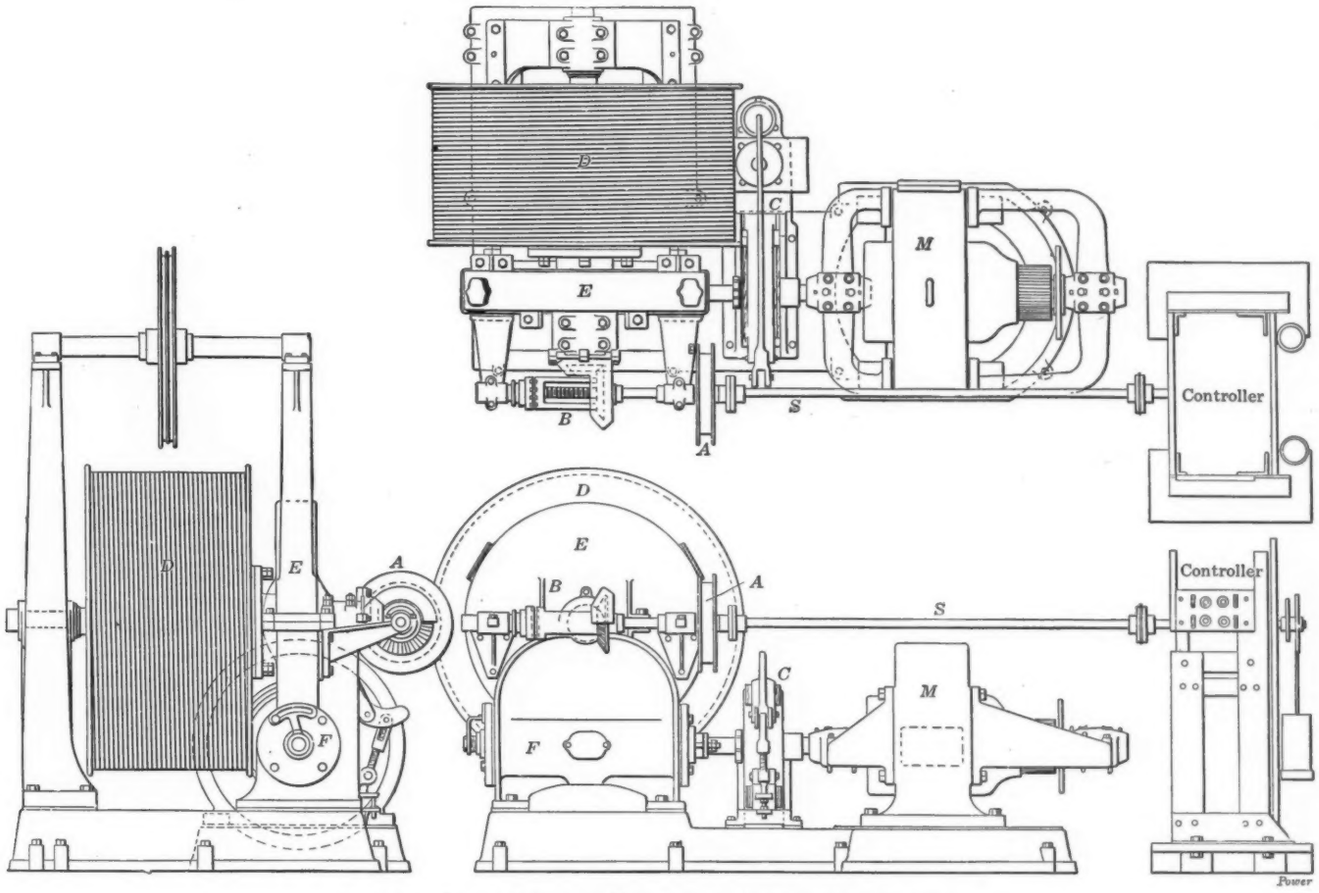


FIG. 115. PLAN, SIDE AND END VIEWS OF ELEKTRON MACHINE

than when resistances are connected in circuit with it, as in Fig. 112. For this reason multiple-speed motors are sometimes made with two field windings connected as in Fig. 113, and resistors connected in circuit with one of these windings. This provides one field winding the circuit connections of which are never disturbed, and there is little danger of losing excitation.

The Elektron mechanical-control elevator shown in Fig. 114 is of the variable-speed type, and its field winding is connected in the manner shown in Fig. 112, with the exception that the speed-controlling resistor, instead of being divided into two sections R' and R'', is divided into a large number of sections, putting the machine in the variable-speed class. Before taking up the controller construction it will be better to describe some of the other details of the machine in which it differs from other machines previously described.

As may be noticed in Fig. 114, the operating sheave A is mounted upon a shaft S that carries the stop-motion mechanism at one end, while the other end extends to the controller and operates the several switches of the latter directly. The rod G is connected with the slack-cable stop so as to throw the brake on if the lifting cables should slack up, and at the same time move a clutch that causes the stop motion to rotate and turn the shaft S, thereby moving the controller switches back to the "off" position. The brake C is of the magnet-operated type. The general location of the several parts of the machine can be more fully seen in the drawing, Fig. 115, which gives an end elevation looking from the drum end of the machine, a side elevation corresponding to the view in Fig. 114, and a plan view. At the end of the worm casing F there is an arm with a long circular slot in its end. This is an adjusting device to take up any slack that there may be in the thrust bearings at the ends of the worm. This view also shows the frames extending upward from the machine to hold the shaft upon which the sliding sheave travels as the ropes wind and unwind on the drum. This is not shown in Fig. 114.

The construction of the brake of this machine is shown in Fig. 116. The magnet M that releases the brake is made so that when energized it lifts the plunger upward, carrying with it the lever D and thereby raising the shoes B, B' from the brake wheel A. The connecting rod D' is made so the distance between the brake wheel and the shoes can be adjusted and the screw D'' is provided to hold the shoes concentric with the wheel. The shoes have leather facings A'. At F is a dashpot to retard the motion of the shoes when the brake is applied and prevent too sudden a checking of the elevator. When the magnet M is deënergized, the weight of the lever D and the other parts hanging on it causes the shoes to close up and grip the brake wheel A.

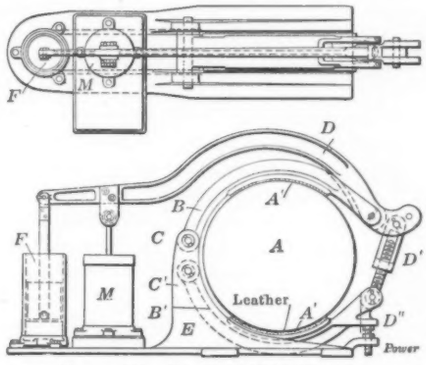


FIG. 116. BRAKE MECHANISM

The controller shown in Fig. 114 carries a reversing switch on the side shown in the illustration, a starting switch within the controller box, and the speed-controlling switches on the back of the controller box. The starting switch is actuated

directly by the shaft *S*, but the other switches are moved by cams mounted on the shaft. The reversing switch is so made that it is at all times in one or the other of the running positions, as may be seen from Fig. 114; hence if the elevator is stopped and then started up again in the same direction as before, the reversing switch is not moved. It is only when the direction of rotation of the motor is reversed that the reversing switch is moved.

The construction and operation of the reversing switch can be better understood from Fig. 117. Upon the shaft *S* is mounted a casting *H* that carries two cams *D* and *D'*. These cams are made of fiber, to insulate the reversing-switch levers *A* and *A'* from the shaft *S*. The cams *D, D'* are not in line, the cam *D'* being nearer the observer than the other. The drawing renders the action of these cams somewhat confusing from the fact that it looks as if the cam *D* were holding the lever *F'* in the position shown by pressing against the roller on the end of the lever. As a matter of fact, however, the roller on the end of *F'* is not in

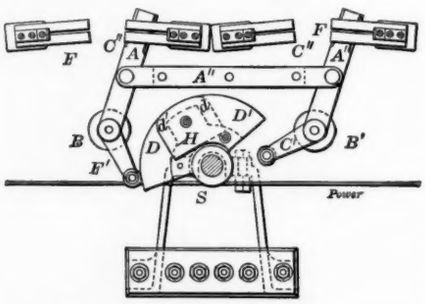


FIG. 117. REVERSING SWITCH

line with the cam *D*, but is in line with the cam *D'*. The lever *F'* was moved into the position shown by the cam *D'*. If the shaft *S* were rotated counter-clockwise, the edge *d'* of the cam *D'* would strike the roller on the end of *F'* and move the latter into the position shown, throwing the switch blades *A* and *A'* into the contact clips *C''* and *F* respectively. If now the shaft were rotated clockwise to the position shown, the friction of the clips *C''* and *F* will hold the levers in the position in which they are drawn. Such a movement, however, would open the starting switch and stop the elevator. If the operator desired to start up running in the same direction, he would again turn the shaft *S* counter-clockwise and close the starting switch without disturbing the reversing switch, but if he desired to start the elevator in the opposite direction, he would turn the shaft *S* clockwise until the edge *d* of the cam *D* struck the roller on the end of the lever *C'* and thereby shifted the reversing switch blades *A A'* over to the opposite side into the other clips *F* and *C''*. The two contact clips *C''* are connected to each other electrically, and the clips *F* are also connected. The clips *F* are connected to one of the armature terminals and the clips *C''* are connected to the other armature terminal, through the starting resistance box. The blades *A* and *A'* are connected to the leads from the supply circuit. Consequently, throwing the blades *A A'* from one side to the other reverses the connections between the motor armature and the line.

Smoothing Commutators

BY JAMES ROCHE

I have often read in POWER of persons grinding down commutators with sandpaper or homemade truing devices, or turning them in a lathe. The latter way is perhaps the only accurate one and is therefore the best, but it is often impracticable on account of the time taken and labor oftentimes needed.

I have found it very convenient and satisfactory to use a piece of grindstone for the purpose. One face of the block is concaved to fit, approximately, the curvature of the commutator, and this curved face is simply pressed against the commutator while the armature revolves. The stone will cut quickly and soon smooth the brush tread of the commutator. Of course, if the commutator is out of true, there is no way to true it except in a lathe or with a truing tool.

A block of grindstone 5x9 inches and 4½ inches thick is in regular use in our station for smoothing the commutators of two railway machines, which are 26 inches in diameter and 8 inches face, a motor-generator commutator 16 inches in diameter and 6 inches face, and for the collector rings on two Bullock and two Fort Wayne alternators. A smaller stone is used for three small commutators.

A spark is hardly ever seen on any of the six commutators, except in case of a sudden overload on the railway machines. All the commutators look like new and as if they had never been touched, and most of them have been in service about seven years in a plant operated twenty-four hours a day. We keep some brushes soaking in oil all the time and keep one of these in each set of brushes, staggering them around the commutator so as to engage the whole face, which keeps it well lubricated.

LETTERS

Care of Rotary Converter Rings

Mr. Eberhardt's question regarding the care of rotary-converter collector rings in the May 17 issue reminded me that there was once a time when my daily bread was earned by making myself generally useful around a substation with six 1500-kilowatt rotary converters. On ordinary loads we found that vaseline applied with a felt swab about once an hour to the commutator and once in two hours to the collector rings gave the best results. When the machines were loaded above the rated capacity it was necessary to grease them a little oftener.

Each week the commutator brushes were carefully inspected and any that were pitted or worn due to sparking were brought into shape again by placing a piece of sandpaper between the commutator and brush and drawing it back and forth, following the curve of the commutator, till the brush was smooth.

At the same time the collector brushes were taken off and thoroughly cleaned and then set back again so as to take care of any unequal wear.

W. L. DURAND.

Washington, D. C.

Motors in Dusty Places

Referring to that part of H. T. Dean's article on "An Overloaded Century Motor" which relates to trouble from wood dust collecting in the sleeve of a pattern-shop motor, I fail to see the necessity of taking this machine down every two or three months to clean out the dust. A far more effective method is to give the motor a good blowing out, preferably with compressed air, at least once a week. Where air is not available, a good hand bellows, vigorously operated, will do fairly well. I should feel exceedingly reluctant to let a commutator type of motor, whether direct-current or single-phase alternating-current, run for two or three months under an accumulation of dust in a wood-working mill, owing to the inflammable nature of this fine dust and the liability of a spark from the commutator setting it off. I have always insisted that motors be cleaned not less than once a week, generally at the end of the calendar week.

A better practice still, where commutator-type motors are purchased specially for wood-mill service, is to buy inclosed motors. Where open-type motors only are available, the writer has got around the difficulty by making two galvanized-iron caps or shields fit the ends of the motor frame, each shield having an oval or oblong hole cut in it for ventilation; the hole was covered with a brass screen or bolt cloth of rather fine mesh so as to exclude the dust. The accompanying sketch illustrates this arrangement. The shield on the pulley end has to be made in two parts, unless the pulley can be taken off without too much trouble and the cap slipped on over the shaft.

On some types of motors, some difficulty is experienced in fastening this dust shield securely, owing to construction of the frame. On most Wagner single-phase machines, however, the shields can be held on by lugs under the yoke-bolt nuts.

I have used this method very effectively on wood-mill motors, as well as on direct-

current machines operating machine tools where there was a liability of metal chips or borings and oil getting into the motor and giving trouble.

Taking apart and reassembling the throw-out mechanism on the Wagner and Century motors is usually such a tedious and unpleasant job that any method which will obviate the necessity of doing so should be welcome.

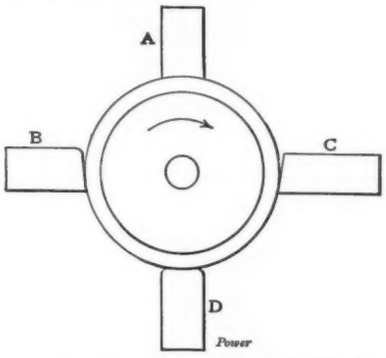
A. P. H. SAUL.

Buffalo, N. Y.

Mr. Lindsey's Burned Brushes

Here is another guess as to the cause of the brushes burning as stated by Mr. Lindsey in the May 31 issue of POWER.

The trouble may be due to any one or a combination of the following conditions: Brush holders unequal distances apart or not parallel with the commutator. That is, the holders at the armature end of commutator may be, say, 14 inches apart while the opposite ends are 13⅞ inches apart.



ONE CORRECT AND THREE INCORRECT BRUSH FACES

The use of too much oil or compound on the commutator.

The brushes may not fit the commutator properly. When fitting brushes to a commutator the sandpaper should be pulled in the direction of rotation, not back and forth. Then, too, if the brushes are quite loose in the holders they should be held firm and as near as possible in the position they occupy when the machine is in operation. The brushes should fit the commutator as at *A*, Fig. 1; the other three are wrong. However, I have seen brushes in one or all three of the conditions represented by *B, C, D,* and if the machine is carrying anywhere near its rated load, there is trouble about like Mr. Lindsey's.

A. K. VRADENBURGH.

Albany, N. Y.

Mr. Lindsey's brushes may be sparking a little because of vibration. If his machine is belted to the engine, the vibration may be caused by a thick spot in the belt, and if direct connected to the engine, lost motion in the crank-pin boxes may cause the vibration.

HERBERT B. BRAND.

Brooklyn, N. Y.

The excessive sparking mentioned by S. G. Lindsey is undoubtedly due to a lack of balance or adjustment somewhere. Machines in normal condition, especially generators of that size, do not act that way unless there is a "screw loose." If I had charge of that machine, my first move would be to see whether the brush holders were spaced equidistantly on the commutator. What makes me suspicious in this respect is the statement that 28 brushes were burned on a six-hour run, indicating that at least some of the brushes are not in their proper relation to the neutral point. The best way to find out whether the brushes bear on the commutator at equidistant points is to raise all brushes and fit a piece of tough manila paper, the width of the commutator, around it, the length of the paper being such that the ends will just meet. Then take a pair of compasses and lay off on the paper, laid out flat, a number of points equal to the number of brush holders. Draw lines squarely across the paper at these points, replace on the commutator and fasten down smoothly with string or a bit of paste under the ends.

Let down the brushes and note if all the brushes "toe the mark." If not, you have in all probability found the trouble, and the brush holders should be adjusted until all brushes line up true to the marks. If, however, the spacing is found all right, and, assuming that shifting the rocker ring to find the neutral point has already been tried, the next move will be to investigate the polarity of the interpoles. As stated in the editorial note, the polarity of each interpole must be the same as that of the main pole next ahead in the direction of rotation. These can be tested out very readily with a pocket compass, which can be bought for 10 cents. However, this latter trouble is rather a remote possibility, but should it exist the polarity of the interpole may be corrected by merely transposing the connections of the coil on it.

Another probable cause for sparking is improperly fitted brushes; they may not touch the commutator across their entire width or face. I have known cases where brushes apparently bore evenly, but upon close examination it was found that only the toe of the brush was in actual contact; consequently the contact surface of the brush was reduced to such a degree that when a heavy load came on, the brush could not carry the current without undue heating, causing excessive sparking and burning of the brush.

I am opposed to commutator lubrication on general principles, and would not permit its use on commutators under my care except in emergencies. There are some rare cases, however, where it may be beneficial. In such cases I would recommend vaseline, but it must be used very sparingly, applied with a piece of canvas or cotton cloth slightly moistened with the vaseline. This treatment sometimes helps to form the chocolate-brown glaze so desirable on a well kept commutator. After this glaze is obtained, no more "dope" of any kind should be used. If graphitized carbon brushes are used, the commutator will take care of itself. That has been my experience.

A. P. H. SAUL.

Buffalo, N. Y.

Identifying the Conductors in a Cable

Mr. Ryan's article on locating faults in circuits, in the issue of May 24, suggested to me that perhaps a method for identifying conductors in a cable already laid, without assistance and with the least possible amount of walking, might be of service to some of the readers of POWER.

Take for example, a 10-conductor cable one-half a mile long. To identify all of the conductors by means of the galvanometer and battery by grounding one end of the conductor would call for a lot of walking back and forth. An easier method would be as follows: At the station end, connect the conductors in groups, each group containing one more wire than the preceding group. Now go to the distant end and with the galvanometer and battery identify the wires in the various groups.

Make a diagram similar to the one here shown. Number the conductors from one to ten and proceed somewhat as follows: Connect No. 1 to No. 9 in group No. 4; No. 2 to No. 4 in group 3; No. 3 to No. 8 in group 4, etc., taking care not to connect more than one conductor of any one group to conductors of any other one group. Indicate on the diagram the different connections and return to the station end where by means of the sketch and the galvanometer, after disconnecting the ends of the various groups, the connections as made at the distant end can be traced out and the station end numbered to correspond with the far end. For example, a wire in group 2 is connected to a wire in group 3. These can be identified and by reference to the sketch the wire in group 2 is seen to be No. 2 and the wire in group 3 is No. 4, and so on.

This method can be used on a cable with any number of conductors and dispenses with the services of an assistant and the necessary means of communication between ends. I would suggest the use of a portable voltmeter, if one is available, instead of the galvanometer.

A. C. KERR.

Ft. Monroe, Va.

Gas Power Department

A New Vertical Engine

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men.

The accompanying engravings illustrate the principal constructional features of the Turner-Fricke gas engine, which is relatively a new comer in the American field. There are no radical departures from modern practice in the design, the aim having been evidently toward improving details of construction rather than the exploitation of any personal hobbies.

As might be inferred from close inspection of Fig. 1, the main framework of the engine consists of a box base and a crank case bolted together, with the cylinders individually bolted to the crank case. The base carries four pillow-blocks on transverse yokes, into which the crankshaft bearings are set, as represented in Fig. 2. The base is ribbed inside and the pillow-blocks are double-ribbed, obviously for the purpose of securing strength without excessive weight. The lower shell of each journal box rests on a massive wedge which is adjusted by means of bolts through opposite sides of the pillow-block jaw (see Fig. 4) to bring the boxes into alinement vertically; they are kept in lateral alinement by the jaws in which they are mounted. These adjusting bolts are reached through hand holes in the crank case, of which there are three, opposite the cranks, in each side of the case, as shown in Fig. 3. The pillow-block caps are fitted to the upper ends of the jaws in absolute line with each other, so that it is necessary only to put the shaft into contact with the caps in order to have it accurately lined up; the caps are lined with bearing metal, like the shells, to permit this contact with the shaft.

The engine operates on the four-stroke cycle, and is therefore equipped with the usual half-speed cam shaft for valve operation—or, rather, with two such shafts, the inlet valves being on one side and the exhaust valves on the other side of the cylinders. The cam shafts are driven from the crank shaft through two-to-one spur gears, which are visible in Fig. 3, and each cam shaft is removable from the crank case, without disturbing anything else, upon unbolting the intermediate journal boxes and taking off one of the end boxes; the latter are cast with flanges covering openings in the crank case large enough to pass the cams.

The valve push rods terminate in the crank case at rocker arms, each carrying a roller in the path of the corresponding

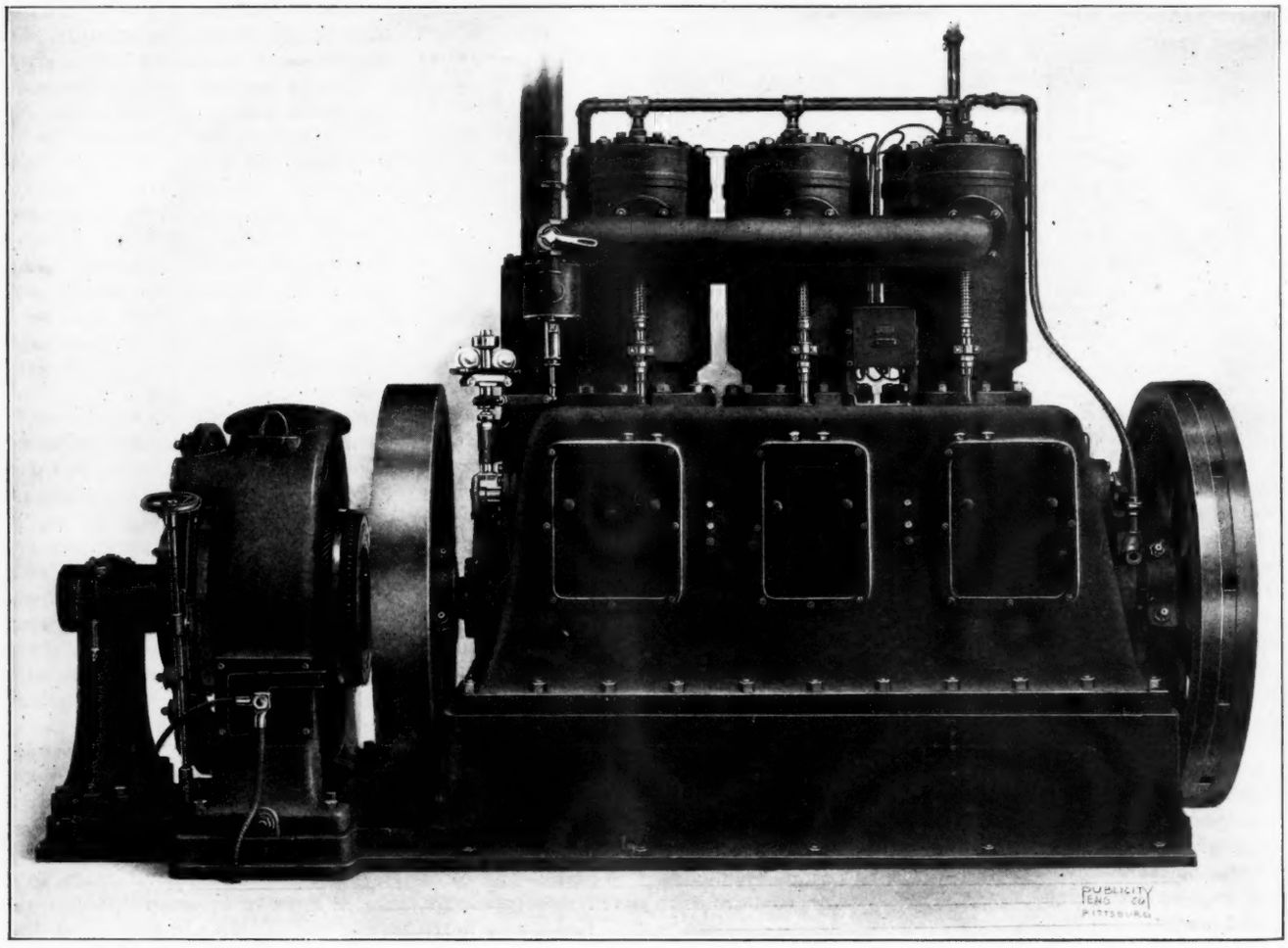


FIG. 1. TURNER-FRICKE GAS ENGINE DIRECT CONNECTED TO A DIRECT-CURRENT GENERATOR

cam. Fig. 4 shows this construction and the arrangement of the valves and other internal parts corresponding to one cylinder. The valves work on removable seats but the complete removable valve cages frequently employed by engine builders are not used, for two reasons: it is easier to cool the cage effectively when it is integral with the main jacket casting than when it is a separate structure set into a pocket, and by providing removable valve-stem sleeves, seats and plugged openings above the seats, all of the operating advantages of removable cages are obtained. Each valve cage is completely surrounded by the jacket water up to the pipe connection. The upper end of each push rod is equipped with means for adjusting the buffer with relation to the end of the valve stem.

The governor is of the simple flyball type, driven through bevel gears from the end of the inlet cam shaft, as shown in Figs. 1 and 5; it controls the port openings of the mixing valve, which is of the multiported piston or "circular grid-iron" type. The proportion of air to gas is adjusted by means of the hand lever, and when once set it remains unchanged by the mixing valve. The action of the valve, under the influence of the governor, is to throttle the mixture according to the load requirements.

The arrangement for adjusting the mixture ratio is unusual and ingenious. The mixing valve *V*, Fig. 6, which is a drum with closed ends, is mounted rigidly on its stem but the stem is separate from the push rod *R* by means of which the governor lifts the valve. Therefore the valve is free to rotate so far as the governor connections are concerned. The bottom head, *H*, of the valve chamber is held on by a circular flange *F* which allows the head to be rotated on its seat; a lever *L*, pivotally attached to the cylindrical spring housing *B*, carries a pin *C* which extends through the chest head *H* into the valve drum; by swinging the handle *L* one way or the other, the valve drum is twisted around so as to increase or decrease the area of the gas inlet openings *G*. The air ports *A* in the valve drum are longer than the corresponding ports in the chest wall, and the valve can therefore be twisted far enough to shut the gas ports entirely without affecting the area of the air ports. The thumb nut on the end of the pin *C* clamps the chest head to the

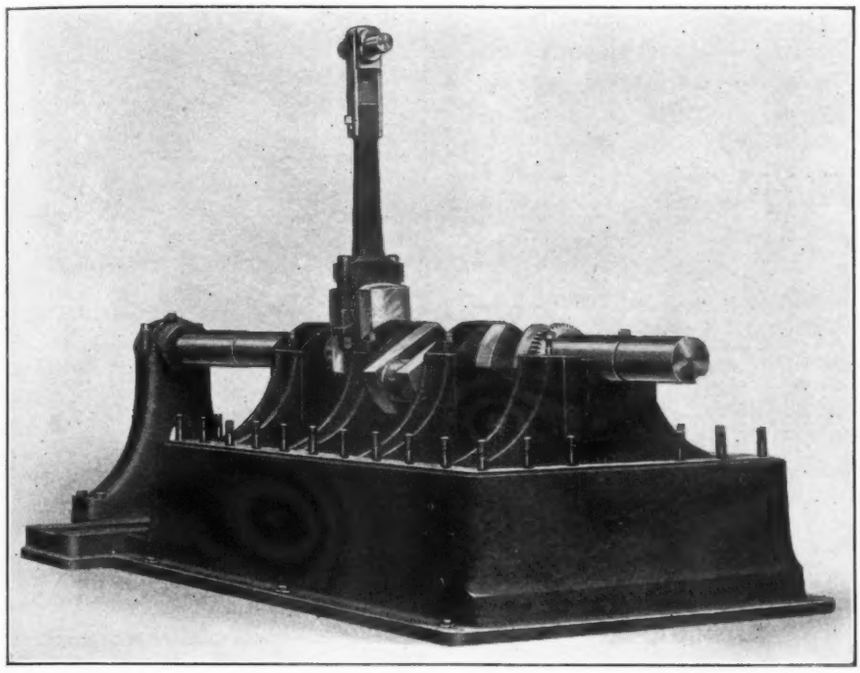


FIG. 2. BASE AND CRANK SHAFT OF TURNER-FRICKE ENGINE

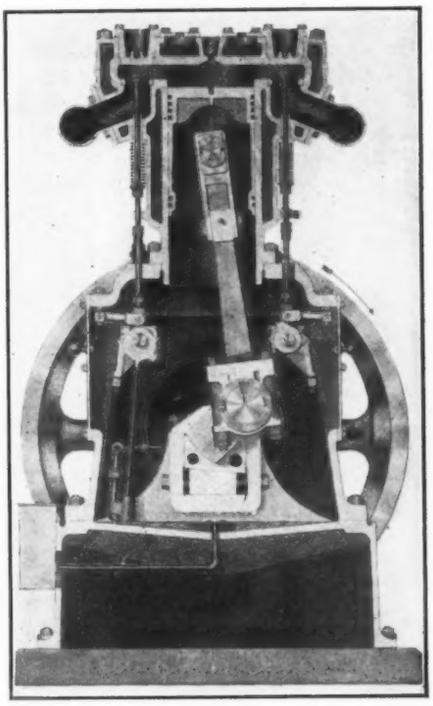
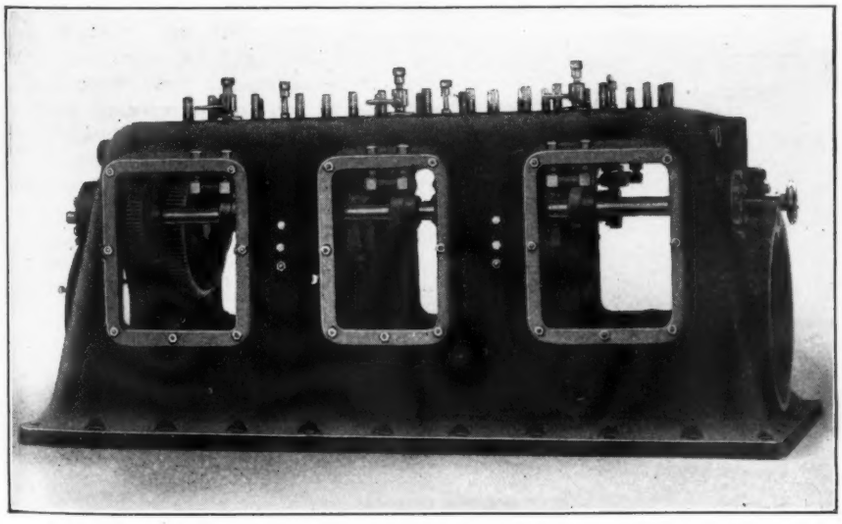


FIG. 4. VERTICAL SECTION THROUGH MIDDLE CYLINDER



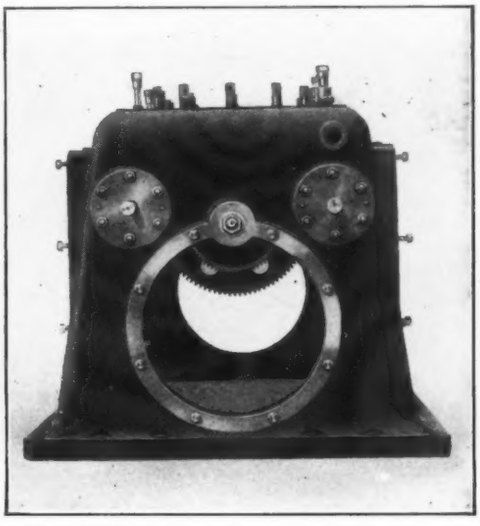


FIG. 3. CRANK CASE, WITH BOTH CAM SHAFTS IN PLACE

flange *F* in whatever position the valve may be set.

The ignition system is of the high-tension or "jump-spark" class, similar to

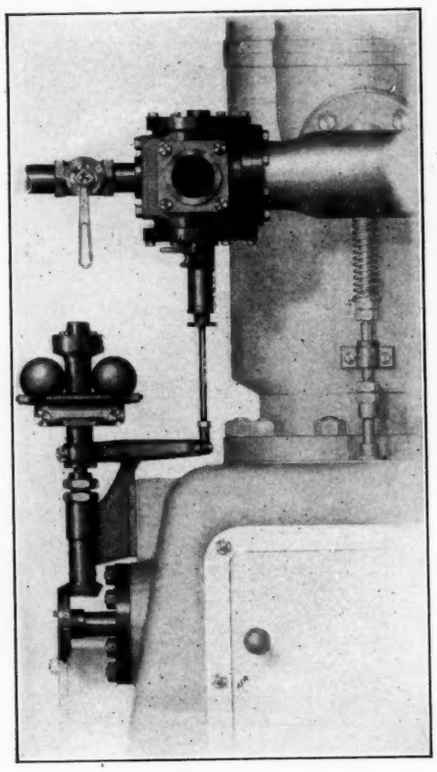


FIG. 5. MIXING VALVE AND GOVERNOR

that commonly used on automobile engines. The spark plug of each cylinder is set in the center of the cylinder head, as shown in Fig. 4, but there is no pocket around the plug on the inside of the cylinder.

cate the crank pins and main bearings. A small oil pump driven by an eccentric on one of the cam shafts (Fig. 4) delivers oil by force feed to the main bearings and crank pins, which are channeled, as indicated in Fig. 7, so that the oil can pass freely between each of the pins and

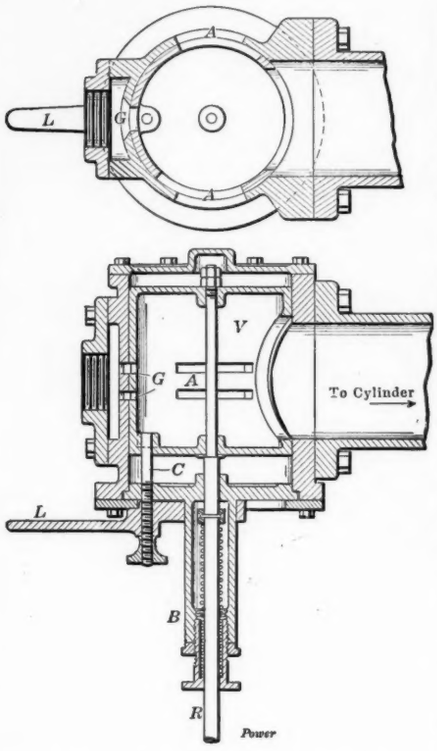


FIG. 6. MIXING VALVE SECTIONS

the two adjacent bearings. The delivery-pipe system is also depicted clearly in this illustration. The pump is shown just

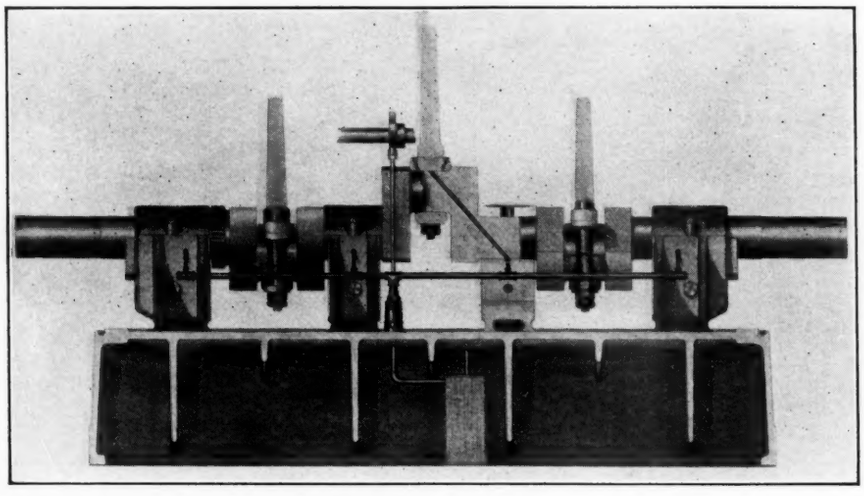


FIG. 7. LUBRICATING SYSTEM FOR MAIN BEARINGS AND CRANK PINS

The pistons are lubricated by the splashing of oil when the cranks dip into oil in the base, in the customary manner, but this is not relied upon to lubri-

above the surface of the base, to the left of the center; it is more clearly represented in Fig. 4.

The engine is started by compressed air through a valve which is put into commission by the admission of air from a storage tank to the valve. The valve is operated by a cam on the end of the inlet

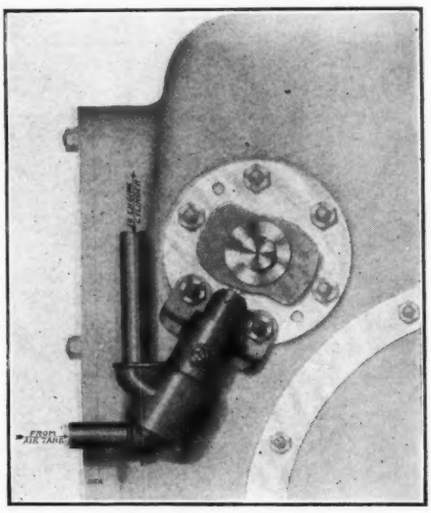


FIG. 8. STARTING CAM AND VALVE

cam shaft, as represented in Fig. 8. When the air supply is turned off, the valve drops out of action.

Another Case of Mere Common Sense

The articles by Frank E. Booth and W. L. James telling about troubles with internal-combustion engines which proved too deep for the local engineers remind me of a case in which my firm sent a high-priced man 200 miles to do a ten second job.

A 10-horsepower horizontal oil engine was provided with a fuel tank in the base plate from which the oil was pumped up to the lamp and vaporizer measure. The engine gave the local force much concern by resolutely refusing to run for more than two or three minutes at a time. They could not get enough oil at the measure so they carefully cleaned the pump-ball valves, blew through the pipes, repacked the plunger, tightened a nut here and slacked one there, to no purpose. He rang up the works and our man went down. The first thing he did, after starting the engine up and having it stop after the regulation three minutes, was to apply his finger nail under fine-wire gauze filter at the bottom of the pump intake and scrap away the waste and filth that had got there by the carelessness of the operator in not pouring the fuel into the tank through the gauze strainer provided for that purpose.

Mr. James speaks truly when he says that this is the sort of talent which is getting gas and oil motors a bad name for "cussedness."

JOHN S. LEESE.

Manchester, England.

Readers with Something to Say

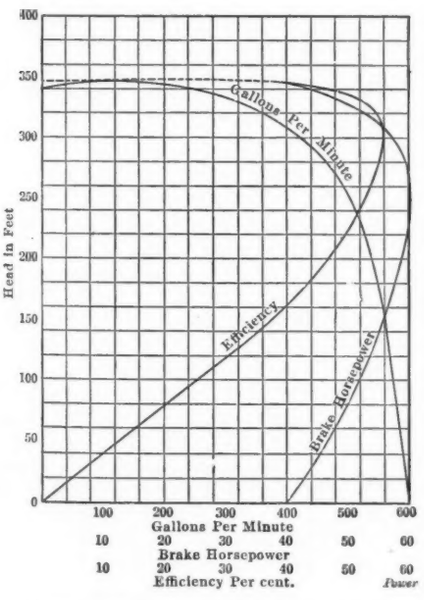
Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted.

Centrifugal Pump Characteristics

The accompanying curves were plotted from data obtained during the test of a 4-inch three-stage centrifugal pump running at a constant speed of 1000 revolutions per minute, and show some interesting characteristics.

The quantity discharged is a maximum at zero head. As the head pumped against increases, the quantity discharged decreases, at first slowly, but at an increasing rate as the head increases. Near the maximum head the quantity pumped decreases very rapidly and reaches zero quantity at the maximum head. Above this maximum head no water can be pumped.

The brake horsepower is a minimum at a head above the maximum head against which the pump will discharge. As the head decreases the pump starts pumping and the brake horsepower rises at once to a value approximately two-thirds of the maximum value. As the head decreases the brake horsepower increases rapidly to its maximum value at approximately three-fourths of the maximum head. Reducing the head further gradually decreases the brake horsepower until at zero head it is again approximately two-thirds.



CURVES SHOWING CHARACTERISTICS

The efficiency which at zero head is zero, rises gradually to its maximum value at about seven-eighths of the maximum head. Above this the efficiency drops off rapidly to the maximum head when, of course, it falls to zero.

These qualities of the centrifugal pump are extremely valuable, especially when electric motor drive is used. They are of the utmost importance when the pump is used for fire purposes, for the breaking of a pipe or hose line does not render the motor liable to burn out or shut down at a critical moment.

WILLIAM SANFORD.

New York City.

A Remedy for Priming

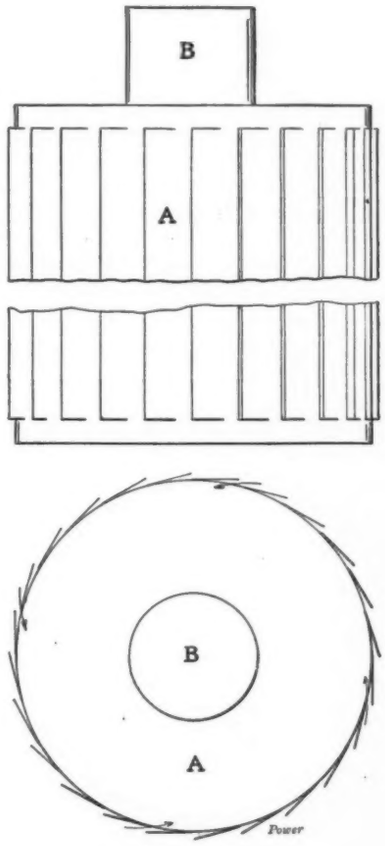
The writer experienced considerable annoyance from priming in the boilers and water being carried over into the engines, until the following scheme was devised, after which all trouble ceased.

The illustration shows the means employed for this purpose. *A* is a cylinder of heavy sheet metal, open at the bottom (although it may be closed if preferred), and at the top is a hole slightly smaller than the steam pipe where it enters the boiler. A flange *B* is soldered to the top of the cylinder for insertion into the steam pipe. Longitudinal slits about 2 inches apart are cut in the metal as shown and are bent outward, forming tangential strips with openings between them of a width of from ⅛ to ¼ inch, according to the size of the separator and the amount of steam which is to pass through.

The apparatus is placed in the boiler with the bottom resting on the flues and the flange *B* projecting into the steam pipe. Its operation is as follows: Owing to the tangential disposition of the long and narrow openings, the steam flows into the cylinder in the direction indicated by the arrows and is thus given a rotary movement at high velocity. This motion causes the particles of water and solid matter to fly outward by centrifugal force, and these coming in contact with the walls of the cylinder adhere to the latter and run back into the water, while the steam, being lighter, passes toward the center and out through the flange *B*.

The flange *B* should be several inches long so as to extend a considerable distance into the steam pipe. The length and diameter of the cylinder *A* must be determined by the size of the manhole and the distance of the steam outlet above the top row of flues. It has been the writer's experience that a very small cylinder will take care of a large volume of steam. In the plant using this device the engine is 26x48 inches in size and runs at 76 revolutions per minute. The separator is 12x48 inches and the openings are ¼ inch in width and 2 inches apart, giving an effective area of nearly 200 square inches. Care must be taken to provide a sufficiently large area for the steam to enter. If the bottom of the cylinder be left open it must be below the water line, otherwise the steam will enter at the bottom and the effect will be lost.

This type of separator has an advantage over those located in the pipe lines, in that it is better to separate the water from the steam before it leaves the boiler, for the obvious reason that heat which would otherwise pass out with the water is thus retained. The use of this scheme has shown a perceptible improvement in the economy of the plant in which it is used and its trial is heartily recommended.



STEAM SEPARATOR

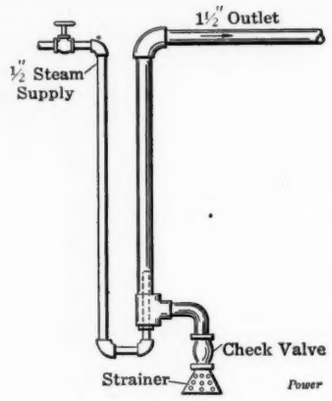
T. TWISTER.

Milwaukee, Wis.

A Homemade Ejector

The accompanying sketch shows a simple homemade ejector which I used successfully to pump out a well with a lift of slightly over 15 feet.

To start the ejector, the steam-supply valve is first opened wide, allowing steam to pass up through the outlet pipe. This creates a suction sufficient to lift the



EJECTOR

water about 5 feet. The check valve then closes due to the weight of the 5-foot column and does not open again until part of this water has been forced up into the outlet pipe and discharged. This operation causes a pulsation about every second, and a steady flow is obtained.

A. C. HARRISON.

Jersey City, N. J.

Troubles with the Tension Carriage

The accompanying sketch shows the arrangement of rope drive in our plant.

We have four 500-horsepower Allis-Chalmers engines, noncondensing and each having a flywheel 14 feet in diameter with 13½ grooves for 1½-inch rope, so arranged that another wheel with the same number of grooves can be added when the engines are compounded. The speed is 100 revolutions per minute and the power transmission is connected to an 84-inch sheave upon a line shaft which can connect two engines together by means of a jaw clutch whenever the conditions of load warrant. The distance between line shaft and engine is 60 feet for one pair of engines and 50 feet for the other pair.

Is the arrangement of the tension carriage and the stationary idler, as shown in the sketch, correct? The tension carriage will not automatically take up the slack in the rope, although there has been as high as 1400 pounds counterweight applied, and it has been found necessary to use block and tackle in addition to the counterweight to pull back the tension carriage whenever the rope becomes too slack.

It is the writer's opinion that the tension carriage ought to be placed next to the sheave on the line shaft instead of the stationary idler. However, there is considerable difference of opinion in this respect. Advice from those who may have had experience with similar conditions or who wish to express themselves would be appreciated.

T. C. THOMPSON.

Chanute, Kan.

Location of Boiler Feed Pipe

Where the feed water should enter a return-tubular boiler and how it should be piped inside are questions which have almost as many answers as there are engineers.

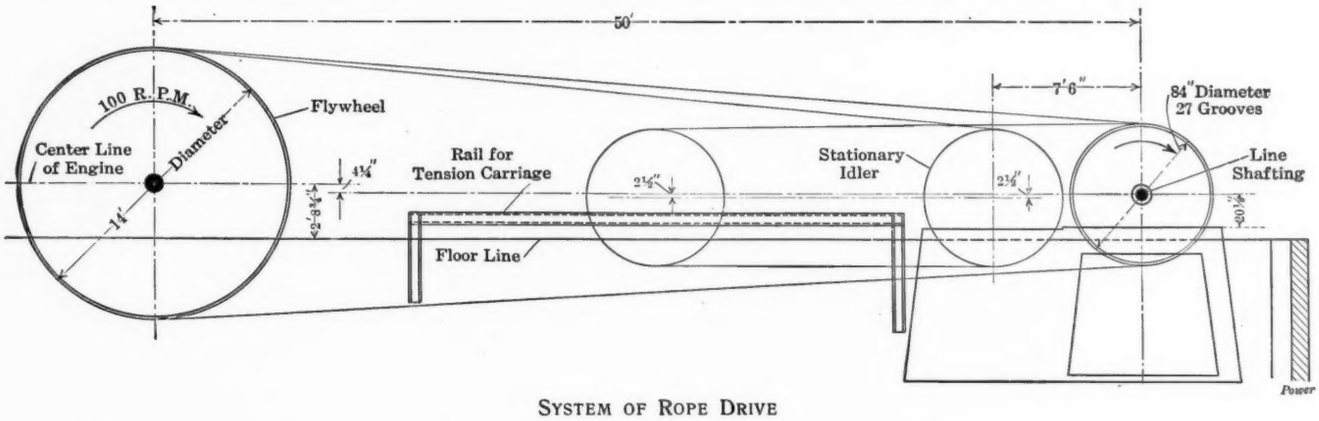
Many advocate entering at one side of the front head above the tubes, then running back at least two-thirds the length of the boiler and across to the other side with an elbow delivering the water between the shell and the outside tubes. The advantage claimed for this method is that the water is heated and prevents unequal expansion of the boiler parts. The economical effect would be "Robbing Peter to pay Paul," and at the best only a few degrees rise in temperature can be secured with only 10 or 12 feet of pipe. This method has a very serious defect, due to the difficulty in cleaning the inside pipes when they become filled with scale, and although it is argued by the advocates of this method that the sediment following the feed water is deposited farthest from the furnace, this would be true only in the case of sand, the usual deposits in the feed water being carried by the circulation inside the boiler and may be left at either end.

A great many boilers are fed through the blowoff. In so far as unequal expansion is concerned this method does not, like the preceding one, impinge the incoming water against the boiler shell, but it does materially assist in the stratification of the different temperatures, and this has a tendency to bow the shell. Nor does it in any way prevent deposits over the furnace. However, in this method the blowoff is protected from being burned, and with a tee located at the entrance to the boiler, the pipe can be cleaned without the necessity of removal.

Again, boilers are fed through the top with a delivery pipe extending down between the tubes to the bottom of the shell. This seems to be the poorest method of all. It causes stratification of temperatures and drives the feed water against the shell with the greatest possible force, sometimes reducing the temperature of the shell and setting up local strains.

This is illustrated in a case observed by the writer, where the feed water was used without being previously heated and in winter its temperature was almost to the freezing point. The boiler had a circumferential crack about 4 inches long over the furnace, caused by improperly driving up a bag, and was plugged with copper rivets. The fireman could not keep an even water level and when it became low, was accustomed to speed the pump to its full capacity until the proper level was gained. The crack was thus kept leaking all the time and the rivets had to be tightened every time the boiler was cleaned.

Another fireman, who was careful to hold the pump at a reasonable speed at all times, took charge and for a period of nine months the crack did not leak. Evidently under the care of the first fireman the shell stood some serious strains which would have been greatly reduced by feeding either through the



SYSTEM OF ROPE DRIVE

front head and across to the side or by feeding through the blowoff.

Some boilers are fed through the front head at one side, and below the top row of tubes. It is about the same as the first method noted, except that the water is not heated in the least before it combines with water already inside the boilers.

Still another method is to feed through the top with the feed pipe reaching just below the lowest water level. A tee is often placed on the end of the feed pipe so as to deliver the water in each direction, and in some instances pipes, drilled full of holes, are screwed into the tee. This insures a thorough mixture of the feed water with that already in the boiler before it reaches any part of the shell. In other cases a pan or plate is set directly under the pipe to bring about the mixing of the water. This seems to be the most reasonable way to deliver the feed water into a return-tubular boiler.

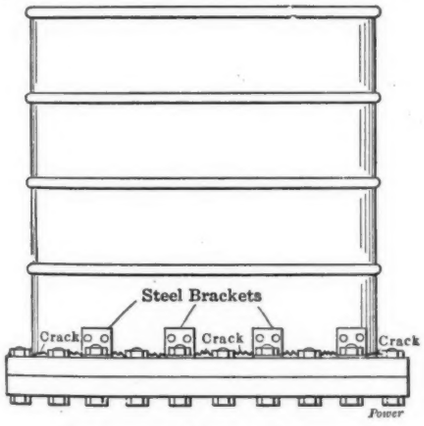
J. R. EARLY.

Columbus, O.

A Repaired Cylinder

The sketch shows a 90-inch compress cylinder which I patched 12 years ago and which is still in operation, and as strong as ever.

This cylinder had a crack extended all the way around just above the flange and caused, as I soon discovered, by tightening too severely on the flange bolts. The crack on one side extended nearly through



REPAIR TO 90-INCH CYLINDER

the metal and the engineer in charge thought that a new cylinder was necessary and that the old one could not be patched.

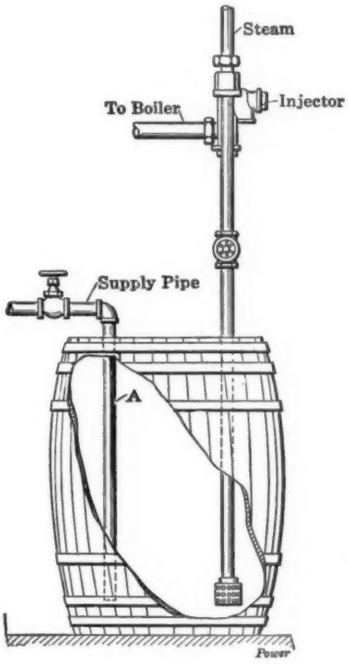
After satisfying myself that the crack had not been caused by the natural pressure which the cylinder was subjected to, and that it had not quite extended through the metal, I decided that it could be repaired. Accordingly I had twelve steel brackets made as shown, allowing the cylinder bolts to pass through the lower flange of the brackets, the other flanges being bolted to the side of the cylinder with 7/8-inch tap bolts. One of these brackets was fitted into every alternate pocket, and when bolted down made a substantial and satisfactory job.

T. K. MCCARTER.

Temple, Texas.

Trouble with an Injector

A few years ago while operating a steam shovel, I experienced trouble with a Penberthy injector in taking water from a barrel, as per sketch. The lift was about 4 feet and the supply pipe ran nearly to the bottom of the barrel. After starting the injector, I made it a practice to turn on the water, so as to



INJECTOR DRAWING WATER FROM BARREL

keep the barrel full, but on so doing, the injector invariably refused to work.

I concluded that the reason for this lay in the fact that the water entering at such a low point in the barrel, set up a swirling motion. I accordingly cut the pipe at the point *A* and did not have any more trouble with the injector.

Coram, Cal. J. H. STRETTON.

Foreign Labor in Steam Plants

Sufficient help, both trained and untrained, is absolutely necessary for the performance or completion of any engineering work. Ordinarily, skilled labor is easily obtained and we find most employers ready to make any reasonable concession in order to obtain the services of men of ability. On the other hand, unskilled labor is absolutely necessary to a greater or less extent in most undertakings, and is permissible from a financial standpoint, so far as the rough end of the work is concerned.

We are being confronted daily with the proposition from the employer, that a certain amount of cheap labor, even in places requiring intelligence, is absolutely necessary in order to keep costs down and thus remain in business. Truly, we have no criticism to offer to any concern or individual for getting things done cheaply, but in the majority of cases is the work done intelligently? Having been up against this proposition for a number of years, the writer has reached the conclusion that the subject is at least worthy of discussion.

In order that any steam plant may operate satisfactorily and safely, men must be employed on whom dependence can be placed. Are we able to draw this grade of material from among this cheap class of laborers? Can the average foreign laborer be depended upon? Do enough of them, even after several years of continuous service, exhibit a sufficient degree of intelligence to make their retention an economical and safe proposition? Do they, as a rule, exhibit that sober, trustworthy and dependable disposition that is so necessary in the operation of a steam plant? What conclusion can we reach? To each question we might answer both *yes* and *no,* with equal candor and truthfulness. They are human and consequently are subject to all the failings of the human race in general, but are they not more subject to these failings than the average untrained American?

It is not the writer's desire to discriminate unnecessarily against persons of other nationalities in any class of work. Some of our greatest engineering achievements are of foreign origin and to these we join in hearty tribute. It is not of this class that I speak. We are called upon to maintain the profession in its high standard, and can it be done under the existing conditions? I have positive knowledge of concerns employing foreign laborers who in the course of a few years have showed enough reliability and intelligence to be placed in charge of large boiler plants. Again, I have had under personal observation, men of this class for several years and find that with the best of treatment and careful instruction, while entirely satisfactory so far as the rough work is concerned in firing, yet they cannot be trusted to regulate the water in the boilers.

Those of you, who for years have been in charge of large plants and have had these fellows to contend with, do you find it a source of comfort and profit? Or addressing the manufacturer and employer in general from a financial standpoint, does it pay and should we be asked to tolerate it? Are they really cheap at any price?

Safety first, economy second. To what extent can we tolerate these conditions and win?

T. M. STERLING.

Middle Branch, O.

Setting Link Driven Valves

In setting Corliss valves, before thinking of lead, compression, etc., we first "square the valves." In setting a common slide valve, the same procedure should be followed. First, the eccentric is placed at mid-stroke; to do this, turn the full side of the eccentric to the extreme inner position and place a mark on the valve stem the width of a rule from the stuffing-box gland. Then turn to the extreme outer position and mark stem as before. Bisect the distance between these two marks and turn the eccentric till this middle mark just touches the rule, and fasten the eccentric in that position; which is just at mid-stroke. Now adjust the valve rod or stem till valve is in mid-position—that is, lapping each port equally. The valve is now squared. All that remains to be done is to put the engine on one center and revolve the eccentric until the proper lead is secured.

Now I should adapt this method for link-driven valves as follows: First, disregarding the position of the crank, drop the link and find the mid-position for the ahead eccentric as outlined above, and fasten the eccentric. Next, lift the link and set the astern eccentric in the same way and fasten it. The relative position is of no account for the present. One eccentric may be up and the other down, or both may be up, or both down. With the link up, adjust the valve to mid-position, thus squaring it with the astern eccentric. Drop the link and see if the valve squares with the ahead eccentric. If it does, the valve is properly squared; if not, adjustments are necessary in the length of eccentric rods. Let these be made and the link raised and lowered again, to test the work. The valve is now squared with both eccentrics. Place the engine on either center and pull the link all the way up and turn the ahead eccentric in the proper direction until the desired lead is secured. Fasten the eccentric there. Now drop link all the way down and advance the astern eccentric to secure the same amount of lead, and fasten it. If the work has been carefully done, the valves are now set. But to test our work, turn the engine to the other center and measure the lead for each position of the link. If a mistake has been made, it may be rectified now by making half the correction on the eccentric rod and half by moving the eccentric.

For instance, suppose we have set the valve on the head end with 1/32-inch lead. Upon testing, the astern position is correct, but in the ahead position we have 1/32-inch lap instead of lead. Any adjustment upon the valve stem would upset the correct setting for the astern position, so we must leave the stem alone. Following the rule, we will adjust the eccentric rod. The total error is 1/16 inch, so by lengthening the eccentric rod the valve is moved 1/32 inch toward the head end. This will bring it in line with the port. It will also obliterate all lead on the head end. Now complete our rule and correct the other half of the error by advancing the eccentric until the valve moves 1/32 inch. This same movement will restore the lead on the head end.

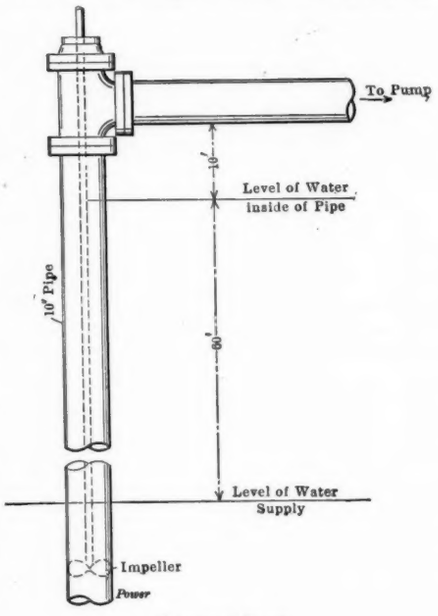
The advantage of this method is that it saves a lot of labor in turning over the engine; also, there is no guessing at adjustments. Whenever adjustment is to be made the precise amount is known. It is the simplest, shortest and best method for a simple slide-valve engine, and, although never having used it on a reversing engine, I cannot see why it is not just as applicable to the latter as to the former.

WILLIAM E. DIXON.

Hudson, Mass.

Will the Pump Raise the Water?

Will some reader of POWER kindly enlighten me upon the following question: Referring to the accompanying



SUPPLY PIPE

sketch, if the water in the pipe is raised to a hight of 60 feet above that of the surrounding water by an impeller or other mechanical means, will a pump situated 10 feet higher, create enough suction to raise the water the remaining distance?

W. F. CALLISTER.

Memphis, Tenn.

Should Superheaters Be Used?

We are investigating the advisability of using steam superheaters in several of our small boiler plants. These plants each consist of from two to four 72-inch by 18-foot return-tubular boilers carrying 125 pounds pressure. Much of the steam generated is carried a distance of from 900 to 1500 feet, and we believe that by superheating enough to give us dry steam at the end of the long lines we could save considerable. We would be very glad to hear from some of the readers of POWER who have had experience with such installations, giving, if possible, a sketch or description of the superheaters used.

A. B. HOLLEY.

Virginia, Minn.

Are the Rim Bolts in Tension or in Shear?

At a plant in this locality a new driving pulley is being installed and is causing much discussion. It is 13 feet in diameter with a 72-inch face and is built up in halves with arms as shown in the sketch. The wooden rim is to be placed after the wheel is in position and will be held to each of the arms by four 1½-inch bolts.

The controversy which has arisen among the engineers is in regard to the stress on these rim bolts. Some say that the bolts will be in shear while others say that they will be in tension. Will some reader of POWER kindly settle the question.

PATRICK MULHAM.

Adams, Mass.

Do Factors of Safety Increase with Age?

It is claimed by some authorities that the factor of safety increases with the age of a boiler. Such an increase would often prove very unsatisfactory to steam users by cutting down the efficiency of their plants, and often making the load too much for the engines.

What I would like to know is, "how much authentic evidence is there to be had in support of the theory that boiler material weakens or becomes fatigued with age?" Of course, by this I do not mean weakening caused by the material becoming thinner.

Has it been proved that the temperature changes to which a boiler is subjected, have any effect upon the strength of the steel? Or, do continual variations in stresses below the elastic limit produce cracks or weaken the material? I should like to see the theories expounded upon this point.

I should not be at all surprised to find that the statistics on boiler explosions show that more old boilers explode, even where inspected, than do new ones, but is this not largely due to other defects than a change in the strength of the steel?

However, I am not opposing the theory but feel that it should be quite well established before any legislation is based upon it.

H. H. HASTINGS.

St. Louis, Mo.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues.

Compression

The interesting article by William Westerfield in the April 26 number prompts me to submit a few experiences.

While it is true that diagrams of certain character indicate certain conditions in the cylinder, it must not be forgotten that of the various types of engines or of valve gears each has a characteristic diagram. A single-valve engine yields, normally, a different diagram than does a four-valve. Yet, under certain conditions, the diagrams of two engines of dissimilar type might closely resemble each

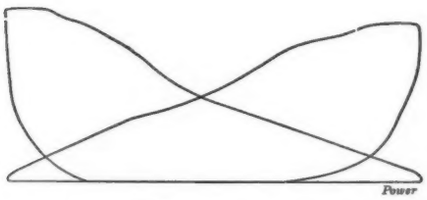


FIG. 1. DIAGRAMS FROM FOUR-VALVE ENGINE

other. In one case the economy might be high, and in the other it might be very low.

The diagrams in Fig. 1, which were taken from a four-valve engine of high economy, resemble somewhat those of Mr. Westerfield's Fig. 3. The engine from which my diagrams were taken is 8x10 inches in size and operates at a speed of 275 revolutions per minute, which gives a piston speed of 458 feet per minute. It is evident from this that this engine is not making its good showing on account of high piston speed. The explanation lies, however, in the fact that it is fitted with Corliss type valves,

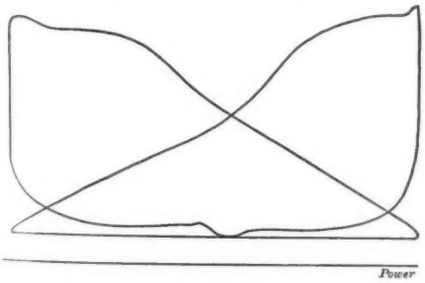


FIG. 2. DIAGRAMS FROM VERTICAL CROSS-COMPOUND ENGINE

which are so fitted in the cylinder that there is but 2½ per cent. clearance.

In Fig. 2, I present diagrams from a vertical, cross-compound engine. The exhaust valves are of the Corliss type while the admission valves are of a multiple-ported plate type. These valves give a very low percentage of clearance. The cylinder is 16x36 inches in size. The engine operates at 160 revolutions per minute under a steam pressure of 160 pounds. This engine shows remarkably good economy. I have not at hand the figures on the water rate, but the coal consumption is only 1.72 per horsepower per hour.

The diagrams shown in Fig. 3 are from an original George H. Corliss engine. They do no discredit to the design. The engine is 22x48 inches in size and runs at a speed of 86 revolutions per minute.

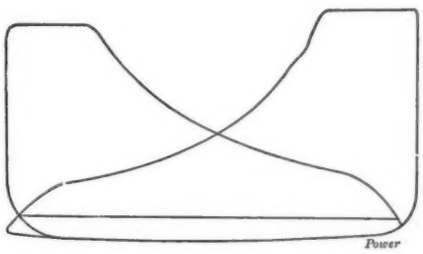


FIG. 3. DIAGRAMS FROM CORLISS ENGINE

The compression is moderately low and the economy good.

In general I have found that with a heavy, varying load too much compression and too quick admission do not produce good economy.

C. R. MCGAHEY.

Sheffield, Ala.

Boiler Blowoff

In the May 31 issue, Milton Heglin replies to G. E. Miles in regard to the above saying that scale lodges under the perforated blowoff pipe, causing bagging and bulging of the sheets in time.

The pipe does not take up much space on the bottom of the boiler, and there is little chance for scale to lodge under it, provided the boiler is not run too long without cleaning. A boiler equipped with such a blowoff would not scale up so quickly as the mud is carried off better. The idea of having just a common blowoff connection is absurd, as such a small hole, 1½ to 3 inches, according to the size of the boiler, will only carry off such mud as settles in or immediately around it.

In my opinion the proper time to blow off a boiler is in the morning before firing up or, as Mr. Miles says, when the fires are banked, as it is then that the water is at rest in the boiler. When the boiler is steaming, the mud is in circulation with the water and blowing off is of little use, unless all of the water is blown out, which, of course, is impracticable.

FRANK GARTMAN.

Sheboygan, Wis.

Why Didn't They Explode?

The first paragraph on the first page of POWER for June 7 is a whole lot said in the nearest way through.

As an inspector of steam boilers with a number of years' experience to my credit, or discredit as the circumstances may warrant, I have often been compelled to admit the truth of the statement that sometimes a boiler will explode for no apparent reason. Again, we can find no good reason why they do not explode.

I was, very recently, called upon to inspect a new "risk" in the mountains of Tennessee. It was a very out-of-the-way place, to reach which I first boarded the "Dixie Flyer," rode a few dozen miles, then took a combination freight and passenger train, rode a dozen or so miles further, then descended to a push car and after a mile or so of this I was confronted with the method of transportation in vogue in Adam's time, so I picked up my grip and tramped a mile further up into the hills to a quarry where the proprietor had a second- (or more) hand boiler on a flat car awaiting my arrival. The first view of the boiler I had was a full-length one with its bottom toward me. I stood still and counted nine separate and distinct patches and remarked on them to the prospective buyer. He replied that he thought those were patches but was not sure. Further and more intimate examination proved the boiler to be practically worthless as a steam generator; it had nearly all of the boiler diseases.

After I had emphatically expressed my disapproval of this boiler the proprietor suggested that I go down to the boiler house and take a look at the boiler then in use but which it was proposed to replace with the boiler I had just inspected. On arriving at the boiler in service, I first noted that the steam gage registered 90 pounds, and as the gage looked new, I surmised that it was somewhere near

correct. The water column and gage glass looked all right and seemed to be properly connected. The flue cap was opened; it was a horizontal return-tube boiler, 60 inches by 16 feet in size. I noticed that nearly all of the tubes were leaking at their front end. The engineer (?) stated that it was impossible to keep them tight because the head was not over an eighth of an inch thick, due to corrosion. The attendant then asked if I would like to look in the furnace and opened the furnace door. On looking in I noticed quite a leak about the center of the first sheet over the fire. As close an observation as it was possible to make revealed the information that there was a longitudinal crack in the sheet about 6 inches long, through which the water was coming in a stream. By questioning the attendant I verified my opinion as to the dimensions of the fracture; then looking at my watch I remarked that I would have to hurry if I caught my train.

Now, in this case there was utter ignorance on the part of all connected with the plant as to the danger they were in. Their idea was that as the boiler was very weak and already had numerous leaks, nothing could happen to it which would be very serious. I am expecting any day to hear that there are a number of the employees of that quarry being picked up in baskets and distributed around to those of their families who care to take the trouble to inter infinitesimal portions of their kinsmen.

My position was such that I could not expand on the merits and demerits of the situation to those intimately exposed to the danger. I did that once in the case of a locomotive and because everybody boycotted that particular boiler on account of what I had said, the owner returned all his boiler policies for cancelation, at the same time exuding the information that he never more wanted a meddling boiler inspector around interfering with his business. Of course, my boss fell on me like a thousand bricks for saying anything to the employees, and the boss was, from a business standpoint, right, but I felt that I owed it to the men who were operating the locomotive to warn them to be ready for a trip to the beautiful shore unless they desisted in their efforts to operate that boiler. Since that time I have grown hardhearted, and besides, I have to look out for my own bread and butter, and I take pains to keep a close mouth to all except the highest in authority; a few more returned policies might cause me to have to look for a new job.

Any boiler inspector can tell that there are often occasions when he meets with circumstances where loss of human life and much property are among the probabilities, judging by the condition of boilers, and we often feel constrained to overstep our instructions and warn the men who would be most likely to suffer personal injury. It is needless to say that in the majority of instances in which a report of such conditions is made to the owner or manager it never gets any further because of the fact that it might interfere with the output of the plant. I once, late on a Sunday afternoon, found a boiler in a sawmill plant in which all the braces in the rear head above the tubes were broken, a crack extended all the way round the segment in the knuckle of the flange, the crack being visible only from the inside. I called the engineer around and sent out for the superintendent who came and took a look for himself and then called the manager in. The latter official said he could not shut down for repairs at that time and asked me for permission to operate the boiler for a few days until some urgent orders could be filled; of course, I could not assent, and was told that the boiler would be fired up next morning as usual; so I immediately made tracks for the telegraph office and suspended the policy on that boiler. The boiler was operated for a week and nothing happened.

I am constrained to believe with the editor that in such instances as the above a watchful Providence alone prevents disaster.

A. C. TERLENE.

Chattanooga, Tenn.

Connecting Rod Crosshead Joint

In reading Mr. Blackwell's communication in the May 17 issue, I am reminded of the experience a friend had with the screwed piston rod and lock nut. A 22x 48-inch Corliss engine was fitted in the above manner and a tram supplied by the makers to adjust the rod to the proper depth in the crosshead. A pound developed in the cylinder and investigation showed the jam nut to be perfectly tight, but an attempted application of the tram disclosed the fact that the tram mark was not in sight. By slacking the nut and turning the rod around till the tram mark was visible it was discovered that the rod had turned in the threads the length of the clearance, allowing the piston to strike the head.

Subsequently, I had an experience with a 16x36-inch Corliss engine which was equipped as above and had a piston with a single ring which had worn a shoulder in the cylinder. A continuous snapping of the ring at one end of the stroke one morning led to an investigation and the same condition was found as in the former case.

These instances prove that a piston rod may turn in the crosshead in spite of a tight lock nut.

In this type of connection the nut, being drawn up, clamps one face of the threads of the rod against one face of the threads in the crosshead, the thrust on the out stroke being taken by the nut while the pull on the in stroke is sustained by the threads in the crosshead. This is an alternation of strains conducive to looseness, while in the split crosshead the threads of the crosshead and rod are tightly wedged into each other, both faces of the threads having equal bearing. In addition, when the pulling apart strain exerted on the rod by the tightened jam nut in addition to the working strain is considered, the split crosshead seems to be the most correct in mechanical principle.

F. C. HOLLY.

Yazoo City, Miss.

Boiler Accidents

The importance of piping a boiler so that nothing can get into it except that which is intended to go in, is often overlooked, as was called to mind by the article, in POWER of May 24, "Soap from Boiler Stops Engine." Fortunately nothing of a disastrous nature resulted on that occasion, but to bring the subject more forcibly to mind I will relate two instances which were disastrous.

A Stirling boiler at Sherman, Cal., several years ago, scattered its various parts over the surrounding country because of an endeavor to make steam from heavy crude oil by putting the oil in the boiler instead of in the firebox. The boiler was equipped to burn crude oil and was arranged to blow the sediment from the oil line with steam. By mistake, the valve was left open or leaked, and the boiler filled with crude oil which carbonized on the shell and tubes. The intensity of the fire heated them to redness, the bottom drum gave way and there was a pyrotechnic exhibition which rivaled Halley's comet as the parts of the boiler, blazing with oil, went flying through the air.

At Sandwich, Ill., about seven years ago, there was installed a return-tube boiler 72 inches by 18 feet in size and having a triple-riveted butt joint. The plant had been shut down for two weeks when one of the boilers exploded, wrecking the plant, killing one man and injuring several others. There had been no fire under the boiler for two weeks. The only explanation was that a valve on the ammonia system either leaked or was left open and the ammonia was allowed to get into the boiler with disastrous results.

While it is well to keep in mind the importance of keeping things from entering the boiler which have no business there, it is of as much importance to be able to get them out when they get there in the ordinary course of events. There are two return-tubular boilers in a plant near Whittier, Cal., which were operated for about five years without having a blowoff pipe attached. The space between

the tubes became packed solid with scale. The tubes had been burned and rolled until the tube holes in the heads were a half inch larger in diameter than the tubes; the heads on one boiler were bulged nearly 2 inches, one being cracked; the shells bagged. New tubes were recently put in with the space around the ends filled with sheet copper and rolled; the bags were driven back. The boilermaker insisted upon putting on a blowoff. Those boilers are being run today—death traps. The boy in charge fired one of those boilers while the boilermaker was there, before it was repaired, and at noon put out his fire, shut off all of the valves, leaving 80 pounds gage pressure in the boiler. When the boilermaker returned from dinner the gage registered 120 pounds. There was a hasty exit of the boilermaker. The boy is not to be blamed for he could not understand how the heated furnace without fire could raise steam. I cannot understand why the boiler did not blow up.

There was, until recently, a small laundry operating near here with the safety valve hanging on a nail in the brickwork of the boiler setting. When asked why it was not in its place on the boiler, the "engineer" said it wasted too much steam so he took it off.

Yet we have no State or national boiler-inspection laws!

FRED W. AXE.

Los Angeles, Cal.

Air Chambers on Pumps

Like Harold James, in the May 17 issue, I cannot understand why more manufacturers do not fit their pumps with air chambers. Again, of those that do, why do some put the air chamber on the discharge, while others put it on the suction? Several years of experience with pumps in the mines of northern Michigan have convinced me that the air chamber is a very important part of a pump. I have had to do with a large assortment of pumps, including single- and double-acting pumps; chest and pot forms, piston or water block, and plunger types; steam, gasolene, air and belt driven; and have yet to see a reciprocating pump that could not be improved, in regards to both action and economy, by a properly designed and located air chamber.

Some engineers and manufacturers will tell you, "Why, we put a relief valve on our pumps and can set it to relieve the discharge of any excessive pressure." You bet they can. But the "Old Man" pays the coal bill.

There is excessive pressure at the start of every stroke that a pump makes. The more speed a pump develops, the more excessive is the stroke-start pressure. Put a suitable size of relief valve on a pump running at its rated speed, and set it low enough to relieve the sudden high stroke-start pressure (which causes water hammer in the valve chambers and pipe fitting) to the same extent that an air chamber would, and it will be found to be discharging almost, if not quite, continuously. It costs money to force water through the resistance of a relief valve. Measure the discharge of a relief valve set to eliminate the water hammer, and find just what that method costs. Then compare this cost, for a period equal to the life of an air chamber, with the cost of a chamber and you will have dollars and cents in favor of the air chamber in an amount that will surprise you, as lots of things do when considered in a financial way.

It is well to consider any mechanical problem, first with regard to safety and second from a dollars-and-cents point of view.

Use a relief valve on any pressure system, by all means, but set it as a safety device and not to relieve part of the working pressure in order to secure smooth running.

By reducing the maximum discharge pressure the air chamber saves money in packing. By running with looser packing there is less friction on the plungers. By minimizing or doing away with water hammer, money is saved in the maintenance of pipe lines and pumps, especially pump valves and valve springs.

Where the water supply is below the level of the pump, and especially when the water is cold, the hammer in the suction pipe is considerable unless the pipe and fittings are of excessively large size.

Although the closer an air chamber is placed to the discharge valves, the more efficient it is, it will give good results in other positions.

If you have a feed-water system that is noisy, or seems determined to leak and blow gaskets, try an air chamber on the pump.

WILFRED E. BERTRAND.

Upper Darby, Penn.

Central Station versus Isolated Plant

In POWER for May 10, W. E. Crane relates his experiences with exhaust-steam heating and states that it was found to be cheaper to use live steam for heating and run condensing, and then states that under such conditions it would be cheaper to buy current from a central station. I am surprised to hear of such a coincident. It would certainly appear plausible, if the exhaust from the engines was entirely made use of to heat the buildings and none of it was exhausted into the atmosphere, that nothing would be lost by such a procedure; also, if we were to change over and run condensing, the same amount of steam would have to be used for live-steam heating as was used for running the engine when running noncondensing. So, in such a case, it would be necessary not only to evaporate enough water to run condensing but also to evaporate enough to run noncondensing. If only a portion of the exhaust were made use of and the balance exhausted into the atmosphere, then under those conditions it might be cheaper to run condensing and use live steam for heating, but whether it would be cheaper to buy current or not would be dependent upon being able to dispense with the greater part of the regular operating force, which is seldom the case as competent men are required to look after the machinery, etc., such as elevators, motors, lights, boilers and piping in case of an office building or hotel.

This section of the country (the Miami valley) is full of paper mills, and not in a single case do the paper machines dry paper with live steam but the exhaust from the engines driving the paper machines and Jordons is used. In mills making a heavy sheet such as boxboard you will find not only the engines driving the paper machines running noncondensing but the main engine as well, which furnishes power for the beaters, etc.

It has been considered good practice to make a pound of paper with a pound of coal in this locality. Recently a mill has been started which is considered to be the model paper mill of the country. This mill runs entirely noncondensing; even the electric-lighting engine exhausts into the main exhaust together with the other engines. In this mill they are making a pound of paper with two-thirds of a pound of coal, which is considered to be extraordinary.

Several mills in the valley have experimented to their sorrow with a central generating plant and motor drive throughout. All of these mills have abandoned the practice.

Mr. Crane speaks of an incident in an office building in which they found it cheaper to buy current and heat with low-pressure steam. I am intimately acquainted with the conditions existing in the larger hotels and office buildings of a neighboring city of some 400,000 inhabitants. In not one single instance have any of them found it cheaper to buy current, nor have any of them been sufficiently suffocated by the arguments set forth by the clever central-station solicitors and power experts, nor bulldozed by the smoke inspector, to a sufficient extent to abandon their private plants and buy current from the central station. I do not wish to imply that they are a bull-headed lot and not willing to be shown, for they have been shown. I know of some hotel people who used central service at 1¾ cents per kilowatt, furnishing their own heat. They finally got wise and put in their own plant and found that they did not burn any more coal during the six months of the heating season than when buying current. also

that the central station would have to supply current at three-quarters of a cent per kilowatt the year round to compete.

The managers of a large office building found that very little additional coal was required during the heating season when operating their own plant as compared to conditions when buying current. In both cases just related, the labor was identically the same as when buying current.

Charles W. Naylor in POWER for May 3 makes the following assertions: "The central station captured the plant by offering a low rate."

How is it possible for the central station to supply current cheaper than the isolated plant, when the vast amount of money invested in transmission lines and underground conduits is considered, also the extent to which the stock is watered and the profits that must be made to pay only a very modest dividend?

"The service has been practically uninterrupted and almost perfectly satisfactory."

Is this saying anything that cannot be said of a modern isolated plant under able and efficient management?

"The central station is lowering the cost of current generation rapidly."

Since when have all the improvements in economy been created solely for the central station? Is it not a fact that the generation cost is only a very small part of the cost of the delivered current, even if sold at the actual cost of production, every item being considered?

Mr. Naylor asks, "Will the isolated plant be able to withstand the onslaught of the central station?" to which I very affirmatively say yes, provided the engineers in charge know exactly how much it costs them to furnish current and heat and elevate themselves to a position of due respect.

Mr. Naylor says, further, that owing to the high land value and the fact that coal must be hauled two or three miles by wagon, also that the sidewalks must be kept clear for patrons' carriages, central-station service is cheaper when it is considered that the noise, dirt, danger and labor of the isolated plant are abolished. If this is so, why do so many of the most recently erected hotels and office buildings have their own plants? Also, why do not the alleged dirt and noise prevent the installation of private plants in such instances in which the most discriminating traveling public are to be dealt with?

As for the sidewalks, I am at a loss to understand how Mr. Naylor's plant disposes of the ash and receives the coal for heating the buildings, which certainly is considerable in so large an establishment. Down in this "neck of the woods" this item has been found to require approximately the same quantity of coal as when using central-station service.

Mr. Naylor speaks of the coal being hauled two or three miles as a good reason for the abandonment of their plant. Why do so many of the large hotels and office buildings find it profitable to haul coal a like or even greater distance?

In conclusion, I would say that the isolated plants of the better class are not "captured" by the central station, for the reason that they can save the owners money. When one is, it is because of the high-class salesmanship (which must also be paid for by the "taken-in" isolated plant) used in hypnotizing the private-plant owners or managers. Again, in quite a number of cases the owners or managers hold stock in the central-station company.

WILLARD BOGART.

Middletown, O.

Operating Costs of Large Units

In the article in the May 31 issue on "Operating Costs of Large Units," the following statement is made in the third paragraph on page 983:

"With eight boilers of 500 square feet of heating surface each, making a total of 40,000 square feet, furnishing steam to one turbine developing 14,000 kilowatts, a kilowatt will be generated by

$$40{,}000 \div 14{,}000 = 2.85$$

square feet of heating surface, or a horsepower by 2.13 square feet, which is considerably less than the 10 square feet allowed by the ordinary boilermakers' rating."

It is undoubtedly true that under the conditions assumed a kilowatt will be generated for every 2.85 square feet of heating surface working in the boiler. Likewise, it is true that this corresponds to 2.13 square feet of surface in the boiler for every horsepower generated. However, this amount of surface per horsepower delivered at the switchboard cannot be compared with the ordinary boilermakers' rating of 10 square feet per boiler horsepower.

A few calculations will serve to show the amount of error. According to the ruling of the American Society of Mechanical Engineers, a boiler horsepower consists of the evaporation of 30 pounds of water at 70 pounds pressure from a feed temperature of 100 degrees Fahrenheit. This is equivalent to 33,305 B.t.u. per hour.

Assuming the same figures as are assumed in the latter part of the article, we find that the heat per pound of steam leaving the boiler is 1290 B.t.u. As we are not considering superheater surface it will be necessary to subtract from this the B.t.u. imparted in superheating. This will amount to

$$150 \text{ degrees} \times 0.60$$

the specific heat of superheated steam which would decrease the total heat per pound of steam to 1200 B.t.u.

Assuming now that the feed water to the boiler comes from an open heater at 210 degrees, we have the total heat imparted to the water per pound by the boiler-heating surface as

$$1200 - (210 - 32) = 1022 \text{ B.t.u.}$$

Dividing this into 33,305 B.t.u. per hour, which is equivalent to a boiler horsepower-hour, we get an amount of steam per boiler horsepower-hour for these conditions of 32.6 pounds per hour.

Using the figure 35,000 pounds per hour per boiler, we find that at this rate of driving, the boiler horsepower would be 35,000 divided by 32.6, which equals 1074 boiler horsepower. The square feet of heating surface per boiler are 5000, so that the number of square feet per boiler horsepower is 5000 divided by 1074, which equals 4.65 square feet. This figure can now be compared with the ordinary boilermakers' rating of 10 square feet per boiler horsepower. Roughly speaking, we would conclude that the boiler is being driven at about 100 per cent. over rating. This rate of driving is very often achieved for short periods, and where economizers are used the efficiency of the whole boiler plant is in no way impaired by the high rate of combustion.

This rate of driving corresponds to about 50 per cent. overload on the turbine, so that when the turbine is operated at its normal load of 14,000 kilowatts, the pounds of steam per boiler would be only 22,750 instead of 35,000 and the square feet of boiler surface per boiler horsepower developed would then be about 7.1 instead of 4.6. Thus, at full load on the turbine the rate of driving of the boilers is but about 40 per cent. over rating.

PAUL A. BANCEL.

New York City.

Receiver Pressure

I have taken great interest in the discussion on the proper compression in steam engines and receiver pressures. It appears to me that the proper compression or the amount of lead to give an engine is not the same for all engines, but must be regulated to suit each individual engine.

The receiver pressure also must be regulated. Each case has its own peculiarities. For instance, two engines are identical in all respects, made from the same patterns, machined by the same men and to the same scale. When sent out they are erected by two gangs of mechanics. The foreman of one gang must have accurate alinement and perfect work. The foreman of the other gang is not so particular, consequently the discrepancy is considerable. When steam is turned on and the engines tested, it is found that one requires more steam to perform the same amount of useful work. The extra steam is eaten up by inferior workmanship and excessive friction.

H. R. BLESSING.

Philadelphia, Penn.

Modern Rotary Steam Engines

By Warren O. Rogers

Many types of rotary steam engine have been designed, but faulty mechanical construction, excessive steam consumption or a combination of both have prohibited their commercial use. Several makes of this type of prime mover are given attention in this article, and a study of their construction will aid the reader in forming an opinion as to their practicability.

In Fig. 1 the engine is shown mounted on a subbase, to which the high-pressure and low-pressure cylinders are secured. Each cylinder is placed midway between two bearings. The shaft passes through the side covers with a free clearance. A cylindrically shaped rotor piston is eccentrically attached to the shaft and the weight of its projecting periphery is counterbalanced, and each engine is counterbalanced within itself. The two cylinders are joined together, and the moving parts as a whole are balanced.

Fig. 2 is a sectional view through the center of the high-pressure cylinder. The shaft, where it passes through the side of the rotor casing, is made steam tight by packing boxes. The rotor piston is mounted on the shaft eccentrically, so that the projecting periphery of the rotor passes the inside wall of the casing with a clearance of less than one-thousandth of an inch on each side. This clearance space is made steam tight by a metallic strip of packing placed in a groove, and pressed outwardly by a flat spring.

Leakage of steam and excessive friction are prevented by a packing ring, contained within a packing ring. The overbalanced periphery is relieved of its extra weight by means of holes passing through the rotor from side to side, which connect depressions on each end of the rotor, and allow any leakage of steam to give equal pressure on both sides of the piston.

The hammer-shaped rider abutment is pivoted at the handle end by a hollow bearing passing through the side casings. This bearing is placed in the direct line of resistance to the steam pressure, with the rider portion between the entrance and the exhaust port. The face of the rider is made steam tight by a metallic packing strip pressed against its vibrating radial surface, by means of a flat spring.

Attached to the rider end is a hardened strip with a cylindrical toe. An intermediate metallic saddle plate, cast in a circular segment, is placed between the rider toe and the rotor. The saddle fits closely to the moving rotor, both being of the same radius, and is held in place by a radial arm hinged to the abutment bearing.

Attached to the handle of the rider is a pivoted pressure rod that enters the hollow end of a pressure piston which closely fits a cylindrical opening, in the outer space of which the steam pressure holds the entire mechanism against the periphery of the rotor.

The saddle has a tongue piece that projects into an opening of the entrance port and holds the rotor packing strip in place. It is far enough forward to form, while in operation, a leverage that prevents the back end of the saddle from being thrown up as the projecting end of the rotor piston revolves. The saddle glides over the surface of the revolving rotor with a rising and lowering rocking motion, while the rounded toe of the rider moves up and down with a constant pressure against the center of the rocking movement of the saddle. The saddle in turn is pressed against the rotor, thereby preventing the steam from taking the short passage from the entrance port to the exhaust port.

The port through which the steam enters is governed by a sliding revolver-barrel-shaped valve operated by an eccentric, which is attached to the main shaft outside the main bearings, thereby admitting steam once at every revolution of the shaft.

This valve closely fits into a cylindrical inner casing having steam ports near the front edge of the valve. The valve has holes passing through it lengthwise, which allows some of the steam to pass through and thus equalize the pressure at both ends of the valve, giving it a free movement while in operation.

This casing is entirely surrounded by live steam under constant boiler pressure. The steam enters the steam chamber behind the rotor, just as its projecting end reaches the bottom of the entrance port. The steam, being prevented by the rider abutment and saddle from passing directly to the exhaust, exerts its pressure in the space behind the rotor piston and forces the piston around. The steam is cut off when the rotor piston is half way to the exhaust chamber, and the expansion of the steam continues to force the rotor to the exhaust chamber, while the momentum continues the revolution of the rotor to the entrance port again.

The steam already in the cylinder continues to expand, and forces the rotor piston around until the expanded steam is released at the exhaust chamber, and then passes to the low-pressure cylinder through the exhaust port. The rotor piston continues its revolution under the acquired momentum, and under the alternate force from the low-pressure cylinder, until it again reaches the bottom end of the inlet port where the steam is admitted

Having done its work in the high-pressure cylinder, the steam passes directly to the low-pressure cylinder, where it again repeats the cycle described for the high-pressure cylinder.

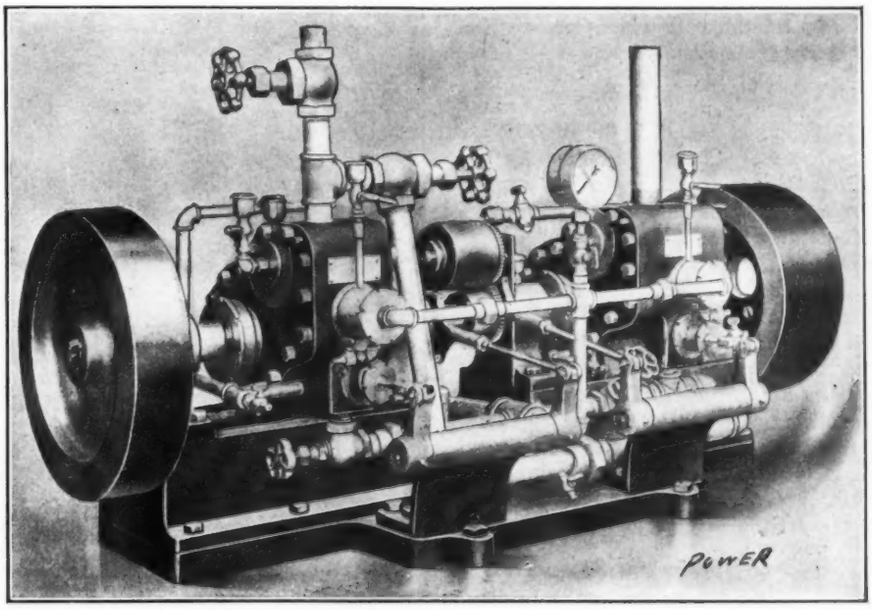


FIG. 1. HARRIMAN ROTARY ENGINE

The interior parts of the engine are lubricated by oil entering with the live steam, and also through the hollow shafting that passes through the handle of the rider abutment. It passes down through holes in the handle of this rider and through the rider itself to the rounded steel toe which presses on the saddle. It also passes through the holes in the hinged arm piece to the saddle. Tests made by Edward F. Miller and Theodore H. Taft show the steam consumption of a Harriman 25-horsepower engine to be 31.02 pounds of steam per brake horsepower-hour. This engine is manufactured by the Harriman Engine Company, 58 State street, Boston, Mass.

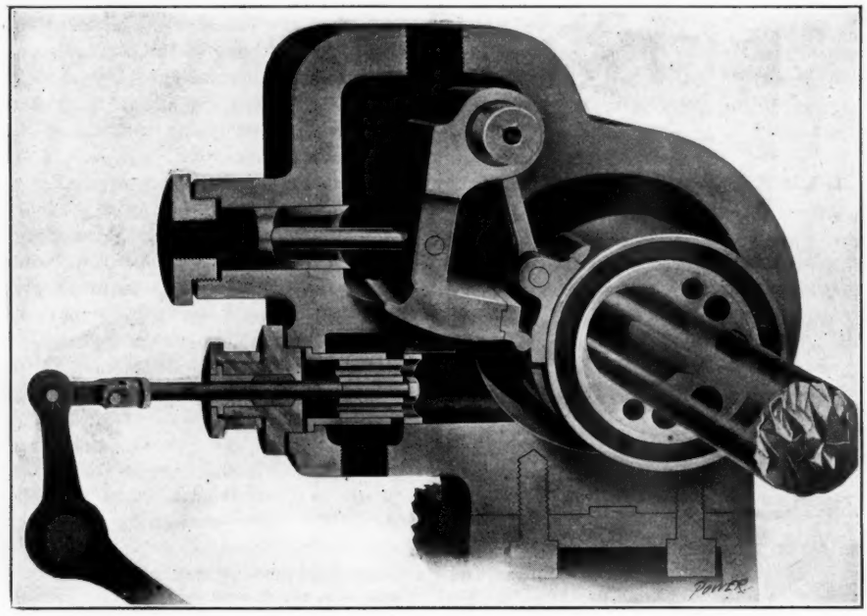


FIG. 2. SECTIONAL VIEW OF HARRIMAN ROTARY ENGINE

KNOWLES ENGINE

This engine, Fig. 3, is the invention of John Knowles, Colorado building, Denver, Colo. It consists of a pair of engines arranged side by side, each being provided with two intersecting cylinders containing rotary pistons coöperatively arranged, and having a driving shaft arranged axially through both the upper and lower horizontal cylinders and pistons of both engines. The pistons and shaft rotate in unison. The engine is also provided with an adjustable rotary cutoff-valve mechanism that will permit the steam to be cut off at different predetermined points or parts of the revolution of the driving pistons in their respective cylinders.

In Fig. 4 is shown a vertical longitudinal sectional view of this engine. Fig. 5 is a transverse vertical sectional view through the rear cylinder, showing the pistons in the position they occupy relative to each other immediately preceding the admittance of steam to the cylinder. Fig. 6 is an outside end view of the engine, showing the valve-governing gear. Fig. 7 shows a sectional view of the steam-inlet shaft, the piston being removed, as at *A;* a transverse sectional view of the same at *B,* and a side and end view of the steam-inlet sleeve upon which the steam-inlet shaft revolves, at *C.* A view of the cutoff valve *D,* which is rotatively journaled in the steam-tight sleeve is also shown.

The operation of this engine is as follows: The steam enters the hub of the forward cylinder head around the sleeve *A,* Fig. 4, through the steam-inlet apertures *B,* into the interior of the rotary valve, from which the steam is discharged through the ports *D D,* Fig. 5, to the port *E* and *F,* of the shaft, Fig. 4, as it rotates. The steam then passes into the cylinder and pushes against the piston arms *G* of the pistons shown in Fig. 5. This causes the two pistons to revolve in their cylinders, the meshing of the gear teeth of the piston causing them to move in unison. The arms of the inlet ports of the valve and sleeve are such, relative to the oppositely proportioned parts of the shaft, that the upper pistons and their shafts have no dead centers when the valve is set to cut off at 70 per cent. of the piston's rotative stroke, as one or the other of the parts of the sleeve and valve are open to the ports of the shaft at all times.

The point of cutoff is controlled by means of an index gear wheel *H,* Fig. 6, which is secured to the outer end of the valve stem. An index plate *J* is secured to the front of the cylinder head. A slidable tooth bar *K* meshes with the gear *H.* The upper end of the sliding bar *K* is attached to a governor which automatically moves the sliding bar up and down as the speed of the engine increases or decreases. This movement is transmitted to the valve, giving it a semi-rotary motion, and in so doing determines the point of cutoff. This cutoff varies from 10 to 70 per cent. of the operative stroke of the engine.

SCHMIDT'S ROTARY ENGINE

A patent for a rotary steam engine has been granted to Rudolph F. Schmidt, 607 East Third street, Cincinnati, O. An end view of the engine, with the cover removed, is shown in Fig. 8. A long-hinged abutment which provides for swift and easy passage of the piston head under the abutment is made hollow on top in order to reduce the weight. The piston wheel is mounted on the main shaft and

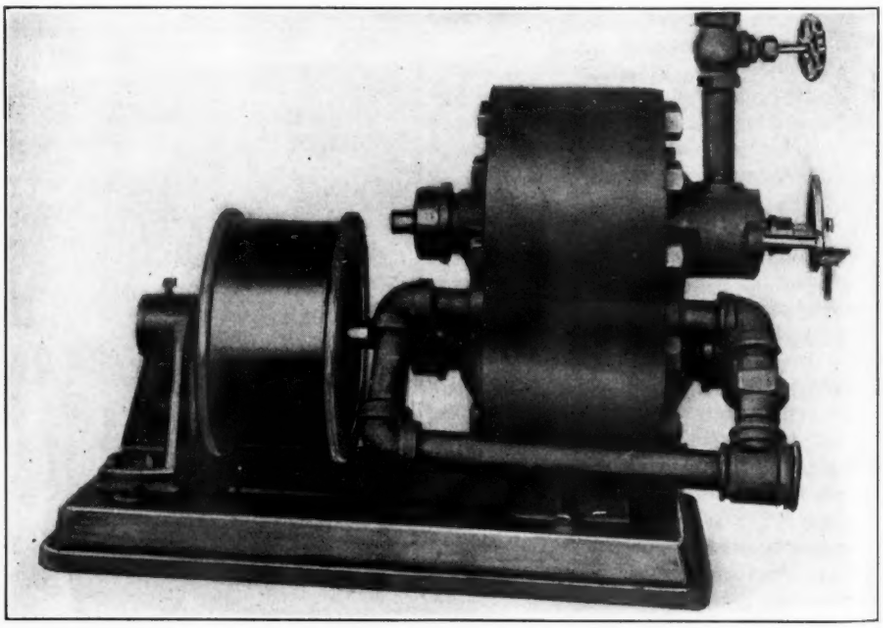


FIG. 3. KNOWLES ROTARY ENGINE

fills the center of the cylinder, leaving a shallow steam channel surrounding the wheel. The piston head is made in two sections, interlocked and curved on both ends, which allows a back and forward movement under the abutment without the aid of cams and levers.

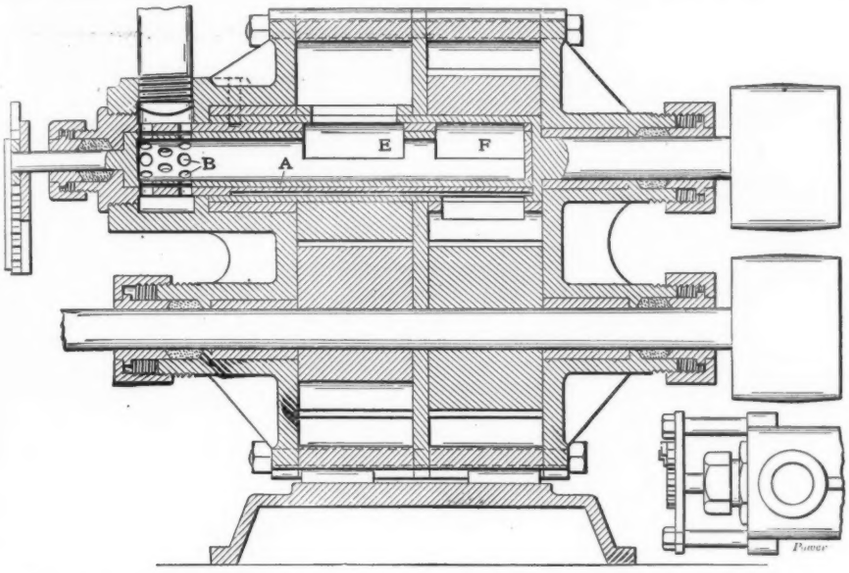


FIG. 4. VERTICAL LONGITUDINAL SECTIONAL VIEW OF KNOWLES ROTARY ENGINE

Split packing rings, which are not supposed to give at high pressure, or cause great friction on the sides of the piston wheel, come level with the rim and partly extend into grooves in the sides of the cylinder, into which they are forced to prevent leakage, and partly into corresponding recesses of the piston wheel, thereby protecting the abutment from wear while the wheel is in motion.

An end spring plate facing against the abutment the full depth of the channel prevents entering steam from passing to the exhaust opening, and a safety spring plate above the abutment counteracts upon it to prevent injury to the engine. The exhaust is always open, thus providing a clear outlet. The drain pipe at the bottom of the casing is to drain the cylinder from condensation before starting the engine.

Fig. 9 illustrates a front sectional view of the cylinder which is composed of three parts: the main body, the cover or cylinder head and the base. The piston shows recesses on both sides of the wheel cover and side of casing, which are for packing rings. Interlocked gearings with set screws on a line with the packing ring and arranged for adjusting the ring evenly into the recesses of the piston wheel are also shown.

Fig. 10 illustrates the cover or cylinder head, showing the levers which counterbalance the weight of the abutment and thus lessens the strain and jar due to the swift contact with the cam. The eccentric for operating the steam valve, and special pin for the bell-crank levers are also shown.

The hinged lever or shifting device is for raising the abutment out of contact with the piston head while it is in motion, which is essential when reversing the engine or the position of the cam wheel.

Each cylinder is equipped as a complete engine, but to eliminate a dead-center position with cutoff at less than one-half stroke, three cylinders are used; with cutoff at one-third stroke, four cylinders, and for full expansion of high-pressure steam, five cylinders are used.

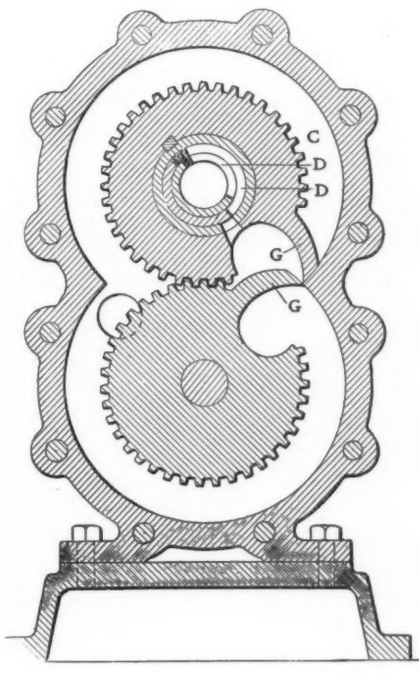


FIG. 5. TRANSVERSE SECTIONAL VIEW KNOWLES ENGINE

The long abutment makes a swift passage of the piston head possible and prevents the steam from passing to the exhaust outlet. The piston head is fastened to the piston wheel, which is keyed to the main shaft upon which, on the outside of the casing, are cam wheel and levers which are attached to the pin of the abutment for the purpose of raising the abutment in immediate advance of the piston head and thus prevent violent contact. There is also an eccentric to operate the valve. The valve does not connect with the exhaust outlet, and allows a cutoff of steam at any point of piston travel, thus utilizing the expansion of the steam. The entering steam, encountering the resistance of the abutment, advances the piston head, and after cutoff the steam is released when the piston passes under the exhaust opening.

A single-cylinder engine of this class requires a flywheel to bring the piston head, when stopped, to a port opening, but with a duplication of cylinders on one shaft, one piston head on each, and each piston wheel arranged to preserve balance and take steam, the dead center is eliminated.

Large port openings, in porportion to the piston surface, are provided, which allows a quick response to the load.

ENGLUND ROTARY ENGINE

According to *The Engineers' List*, this engine is the invention of Albert O. Englund, Wayne, Kan. In its general features, the engine comprises a fixed annular cylinder having a piston moving therein, which projects outward from a central shaft. Within the cylinder are one or more gates which, when closed, form

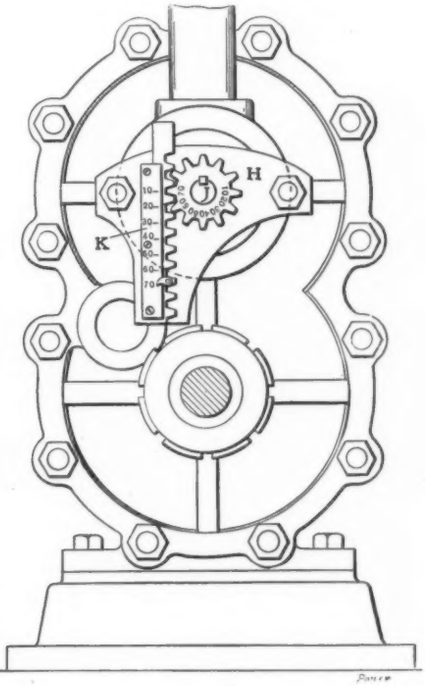


FIG. 6. END VIEW OF KNOWLES ENGINE

abutments, or cylinder heads, and which, when open, permit the passage of the rotary piston and then close.

As will be seen from Fig. 11, the cylindrical cylinder shell is mounted on a base

provided with suitable bearings for a central rotary shaft. Projecting outward from the shaft is a collar which carries the piston disk.

The cylinder is slotted on its inside circumference, and the collar on which the disk is mounted fills this slot while suitable packing rings prevent the passage of steam out through them.

At diametrically opposite points on the cylinder are located gate chambers, containing the gates.

Communicating with the gate chambers on either side of the gate are steam-inlet and exhaust valves. These are operated by rock shafts, see Fig. 12, which are actuated by inwardly projecting curved arms, having antifriction rollers at their ends. These are engaged by cams mounted to rotate with the central main shaft, but so mounted that they may be shifted with relation to each other to control the time of opening and closing the inlet and exhaust ports. The lift of these cams, and the initial opening of the valves are accomplished by means of a hand lever, engaging with a movable sleeve extending across the cylinder. These gates each consist of two semi-disks, each half-disk being mounted on the end of a curved arm, which in turn is mounted on and turns with a vertical rock shaft. The two rock shafts which together actuate the two halves of the gate are geared by sectors to move together though in opposite directions. Projecting rearward from these rock shafts are arms which in turn are connected by a system of levers and rocking bars which are acted upon by a cam rotating with the central shaft. As the shaft and cam rotate under the action of the circularly moving pistons, the gates on one side or the other are opened to allow the passage of the piston, or closed after it to form the cylinder head.

Mounted on this central shaft is a pulley which, by means of a belt, drives a governor for controlling the inlet of steam to the cylinder.

When steam is admitted to the cylinder, it passes into one of the valve chambers and through one of the partly opened gates, the other gate being closed. The steam impinges on the rear face of the piston driving it forward. As the piston travels forward, the gate behind it closes, and an inlet valve opens just forward of the closed gate, while the exhaust valve of the gate ahead opens. This valve closes just before the piston reaches it, and the second or right-hand gate opens. After the piston passes the open right-hand gate and the inlet port of that gate is opened, the right-hand gate closes. After the piston has passed the right-hand gate and the inlet behind it has opened, the left-hand gate closes and the left-hand exhaust is opened before the moving piston.

In the first half of the revolution, the left-hand gate closes shortly before the right-hand gate opens and in the second half the right-hand gate closes shortly

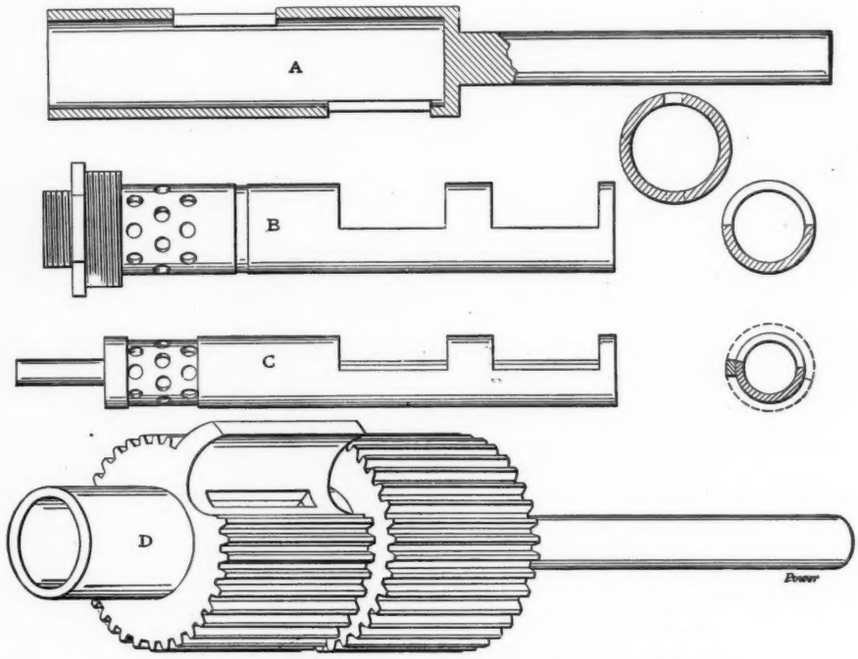


FIG. 7. DETAILS OF VALVE AND ROTOR, KNOWLES ENGINE

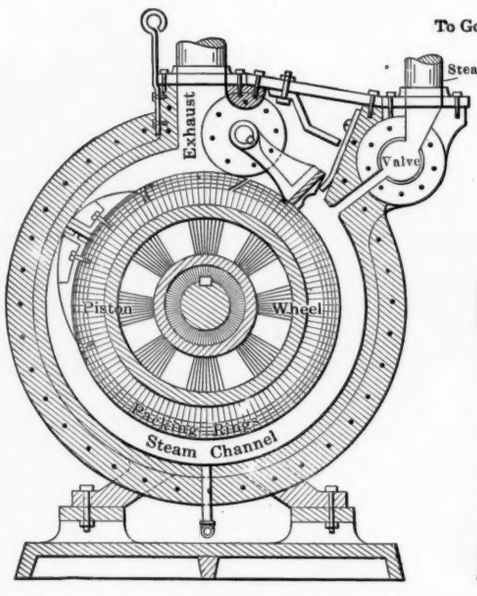


FIG. 8. SECTIONAL VIEW OF SCHMIDT'S ENGINE

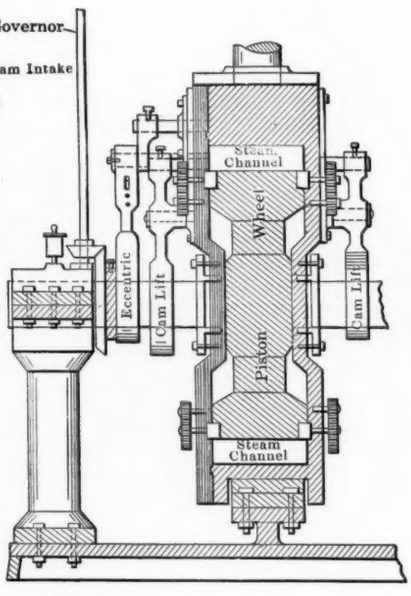


FIG. 9. FRONT SECTIONAL VIEW

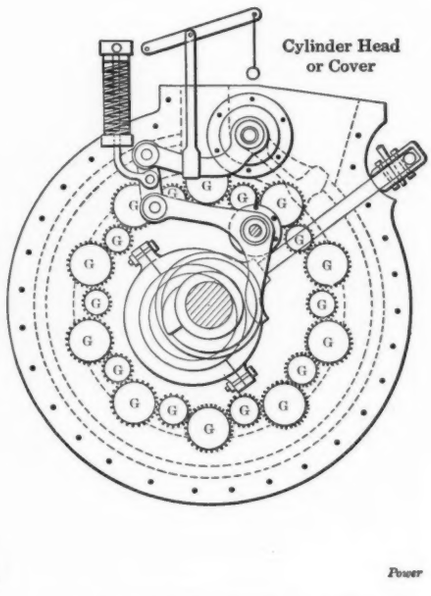


FIG. 10. EXTERIOR VIEW OF CYLINDER HEAD

before the left-hand gate opens; thus each gate closes with pressure on both sides. While the gates are opening they are not under pressure.

If the load on the engine does not require the piston to operate under a direct head of steam during its whole travel, the engineer moves the cam-adjusting lever rearward, and thereby holds the inlet port valves closed for a longer period than when the steam is not being used expansively. All the valves are of rotary type and are actuated by cams. The

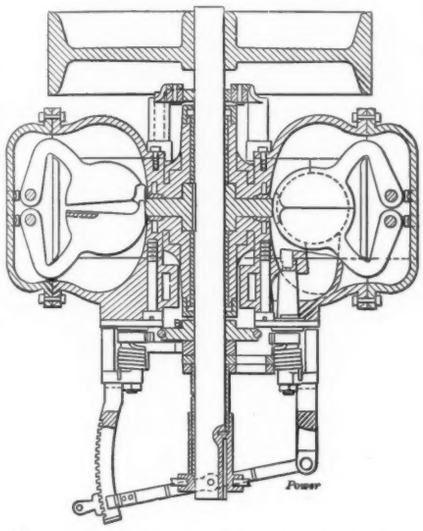


FIG. 11. SECTIONAL VIEW OF ENGLUND ROTARY ENGINE

two sides of the gate move in gate chambers or slideways which hold them rigidly in alinement with each other.

MOTSINGER ROTARY ENGINE

This engine is shown in Fig. 13 direct connected to a small direct-current generator. The details of construction are shown in the accompanying illustrations.

The engine consists of three cast-iron rotors, as shown in Fig. 14. The center rotor is made with a round groove, and the two outer rotors each have a projecting tooth with a round face, which fits into the groove of the center rotor. These rotors are connected by means of gear wheels which are secured to one end of their shafts and cause the middle rotor to turn at such a speed as to permit the tooth of each outer rotor to fit in the groove. By setting the rotors 180 degrees apart, the vanes of both outside rotors may alternately use the groove in the middle rotor.

These rotors are incased in a cylinder having three bores whose centers are in the same horizontal plane. The radii of the outside bores are as much greater than the radius of the middle bore as the hight of the vane; the outside bores cut into the middle bore a depth equal to the difference of their radii. Placed in proper bearings of the heads covering the ends of these bores, the three rotors lie in contact with each other their entire length, forming two steam-tight chambers swept by the rotating vanes on each side of the center rotor. Both heads are cast from the same pattern and their steam chests each contain three chambers communicating with the ports that lead to and from the engines.

Fig. 14 shows the arrangement of the three rotors, also steam ports.

The tops of the rotor vanes are made steam tight by fitting T-shaped packing strips into T-shaped grooves, allowing a little space at the bottom for movement and wear. Small coil springs inserted beneath these strips and the centrifugal force of rotation hold them tight against the surface of the cylinder. The middle rotor is packed in the same way by the use of four packing strips set 90 degrees apart so that there are always two of them (one above and one below), doing service in preventing steam from passing from one engine to the other. The ends of the cylinders can be similarly packed, but as the ends of the rotors fit snugly against the heads, and there is so little wear, it is said that there is no need for packing. To take up the wear of the bearings and prevent any possible leakage between the rotors into the exhaust opening, a spring gib is used against a split bushing. It is adjusted and held tight by a set screw and jam nut against each bearing of the outside rotors.

Since both heads are alike and face each other on the engine, the positions of the ports *A* and *B*, Fig. 14, will be reversed, and when the head, shown removed, is put in place, if one could look through the engine toward the flywheel, the far port for the left-hand engine would be exactly opposite the upper port *X*. The left-hand rotor is shown in position to take steam at the far end of the cylinder, and when it has made almost a complete turn will exhaust through port *B* of the near head below the plane of the centers of all the rotors. Looking through the right-hand engine one would see the far port opposite lower *X*, and the right-hand rotor would be shown as having gone half a turn since taking steam and would exhaust through the near port

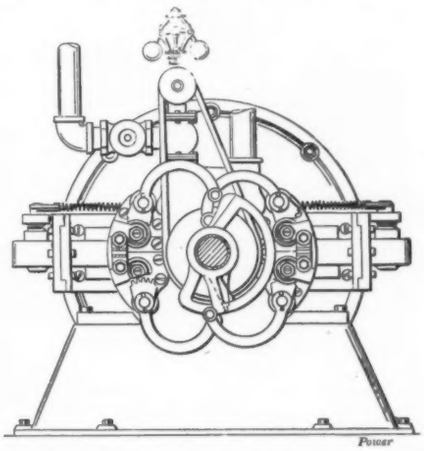


FIG. 12. END VIEW OF ENGLUND ROTARY ENGINE

A, above the plane of centers of all the rotors.

Fig. 15 is a sectional view through the center line of the engine, showing the design of the valve and the passage for the steam to the exhaust.

The governing device, Fig. 16, consists of a central stud on which a gear revolves, driven from a second gear secured to the shaft of the center rotor. To the top gear are hung two weight arms, which are kept in position by means of a spring fitted to each. To the pivoted end of each weight arm is secured a segment and gear wheel which mesh with a gear fitted to the valve stem of the engine. As the speed of the engine increases or decreases the governor weights are moved in or out, and the gear segments, moving with the weight arms, turn the valve which cuts off the steam, earlier or later, as the speed of the engine dictates.

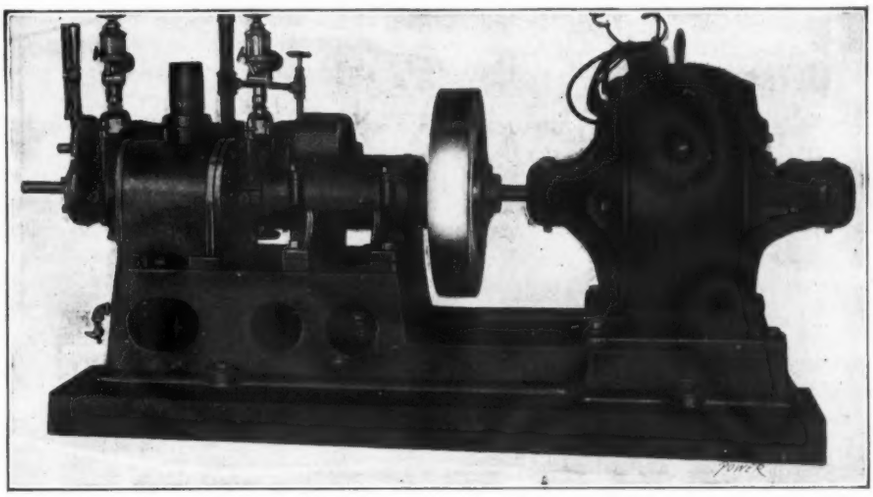


FIG. 13. MOTSINGER ROTARY ENGINE

Steam enters the engine and passes to the port that registers with the two middle ports of the reverse valve shown at *A*, Fig. 15. Directly opposite and below in the live-steam port is the exhaust port. It also registers with the middle port of the reverse valve. The middle ports of the reverse valve are separated by a partition which makes the port on one side communicate with one head, and the port on the opposite side communicate with the other head, so that when the port is open for live steam to enter one end of the engine, the other port is open to receive the exhaust from the other end. The inlet and exhaust ports are on the same side of the valve but at opposite ends. The exhaust ports always open both behind and in front of the cutoff plate, making a free exhaust possible.

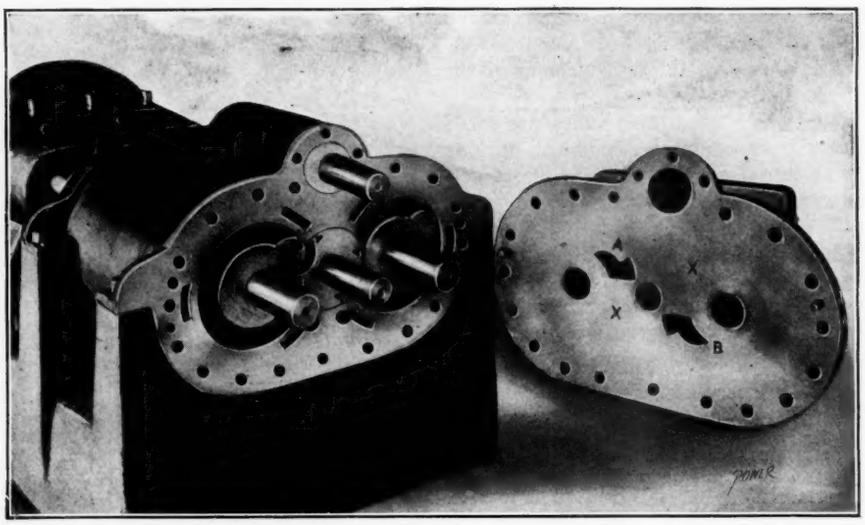


FIG. 14. VIEW OF ROTORS AND PORTS IN CASING, ALSO PORTS IN HEAD

At the near end of the valve, Fig. 14, are shown three inlet ports. The largest port opens into the steam chest in front of a cutoff plate, and steam coming through it is fed to each engine alternately by the port in the revolving cutoff plate. The two slot-like ports open behind the cutoff plate to both engines where there are no dead points. The ports in the heads being on opposite sides, the engine is reversed by a half turn of the valve handle. When the reverse handle is perpendicular all steam is shut off both engines, but if moved either way to a point within 15 degrees of the horizontal, the two ports behind the plate will open to both engines where there are no dead points and the engine will start with any load it can pull. Pushing the handle completely home to its resting lug, the ports beneath the plates will be closed and the engine will run on the "economic" cutoff plates.

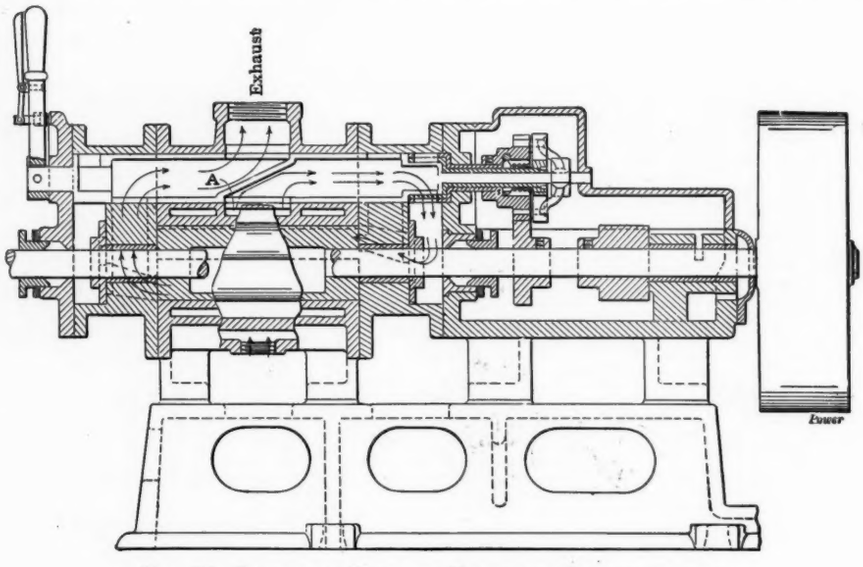


FIG. 15. SECTIONAL VIEW OF MOTSINGER ROTARY ENGINE

The exhaust is always open in front of the steam-driven vanes. The ports in the heads, Fig. 14, are cut the shape of a stylish shoe, *A* inverted and *B* upright. The toe part of the shoe is the main exhaust for the steam behind the vanes, but the heel part opens into the groove of the middle rotor, and as this groove naturally exhausts first, this heel part of the port serves to admit steam at the earliest point desired when running one way, and to let out the last bit of water or steam contained in the groove as the vane dashes quickly by in the opposite direction.

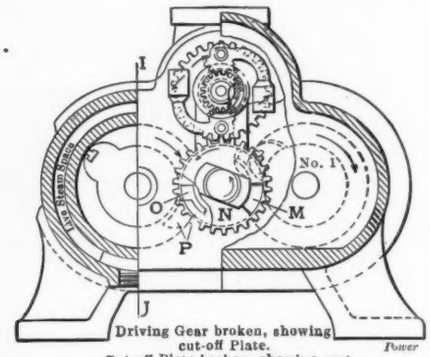


FIG. 16. GOVERNOR AND VALVE GEAR

This engine takes steam once every revolution, and exhausts but once. Steam pressure is maintained during almost a complete revolution.

The equal circular bodies of the rotors are cored to balance by taking out metal from the side the vanes are on and leaving in metal on the grooved side.

This engine is made by the Motsinger Rotary Engine Company, Greensburg, Penn.

The weight of the dry matter in a cubic foot of wet mud of the density to be expected at the bottom of a reservoir was determined by the engineers of the Tallahatchie drainage district, Miss., in connection with their investigations of the probable sedimentation in the reservoirs that may be constructed in the district. Two sections, 8 inches long, of 3½-inch wrought-iron pipe, were turned on a lathe to a cylinder bore inside, were milled at the ends, beveled outside at one end, and carefully measured with calipers and scale. According to the *Engineering Record,* each tube was pushed down into typical submerged mud, struck at the ends and the contents then washed out into an iron vessel, which had been first heated to drive off moisture, and then weighed. The vessel was then heated until all moisture had been driven off, and again weighed. By several such determinations it was ascertained that 1 cubic foot of wet mud, such as was tested, contains 60 pounds of dry matter, and "parts by weight" of sediment can be readily reduced to cubic yards of wet mud which would deposit in a reservoir, provided the column of water passing through the reservoir be known.

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
150 Michigan Avenue, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for June, 1910, *was* 139,000.

July 5..... 34,500

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

Contents

PAGE

Hydraulic Turbines for 670-Foot Head 1192
Applying Patches to Boilers..... 1195
Central Station Steam Heating..... 1198
Factors Affecting Cost of Heating..... 1200
The Operating Engineer's Problem..... 1201
Hydraulic Rams and Their Capacity... 1202
The Electric Elevator..... 1204
Smoothing Commutators..... 1206
Care of Rotary Converter Rings..... 1206
Motors in Dusty Places..... 1206
Mr. Lindsey's Burned Brushes..... 1207
Identifying the Conductors in a Cable 1207
A New Vertical Engine..... 1208
Another Case of Mere Common Sense 1210
Practical Letters:
Centrifugal Pump Characteristics ....A Remedy for Priming....A Homemade Ejector....Trouble with the Tension Carriage....Location of Boiler Feed Pipe....A Repaired Cylinder....Trouble with an Injector....Foreign Labor in Steam Plants....Setting Link Driven Valves....Will the Pump Raise the Water?....Should Superheaters Be Used?....Are the Rim Bolts in Tension or in Shear?....Do Factors of Safety Increase with Age? 1211-1214
Discussion Letters:
Compression....Boiler Blowoff.... Why Didn't They Explode?.... Connecting Rod Crosshead Joint ....Boiler Accidents....Air Chambers on Pumps....Central Station versus Isolated Plant....Operating Costs of Large Units....Receiver Pressure 1215-1218
Modern Rotary Steam Engines..... 1219
Editorials 1225-1226
Chimneys and Draft..... 1227
Hoisting Engineers' Examination..... 1228

Power and The Engineer

Editorial

The Hydrostatic Test

There seems to be a general impression among laymen and those associated indirectly with power plants, that if a boiler shows no signs of distress under a hydrostatic pressure of a certain number of pounds, it is, therefore, capable for the next twelve months of carrying with safety a steam pressure equal to two-thirds of the hydrostatic pressure applied; or, to put it in another way, if the boiler does not rupture or show signs of distress under a water pressure of 150 pounds, it is safe to carry a steam pressure of 100 pounds. This is very illogical, and one might as well say that because a man can lift 150 pounds six inches from the floor—once, he ought to hit a blow of 100 pounds all day long for the next twelve months. We are aware that the comparison is a little overdrawn, but there are times when only a strong contrast will bring out vividly a commonplace truth.

Suppose a boiler with an ultimate bursting pressure of 400 pounds. Using a factor of safety of 5, the allowable working pressure would be 80 pounds, and 1.5 times 80 pounds would be 120 pounds, which would be the logical hydrostatic test to apply. As the elastic limit is generally a little above one-half the ultimate breaking strain, this boiler would stand a water pressure of, say, 200 pounds without stretching the metal beyond the elastic limit. Suppose such a pressure were applied, and then that pressure were divided by 1.5 to obtain the steam pressure. The result would be 133 pounds, giving a factor of safety of only 3, which, of course, is dangerously low. The boiler would appear to act about the same, whether it was subjected to a hydrostatic test of 120 or 200 pounds. In one case, the allowable pressure would be 80, and in the other, 133 pounds, such figures, of course, being meaningless until the accompanying factor of safety is determined.

Not only may the boiler be strained beyond its elastic limit by the application of the hydrostatic test, and a permanent set take place, but a boiler almost wholly void of ductility would show no signs of distress under the application of a steady hydrostatic load, whereas such a boiler might fly all to pieces when subjected to the alternating stresses produced in it when under steam.

That there is no relation between the hydrostatic test and the steam pressure to be allowed, is shown by the lack of uniformity on this question among different governing bodies. While the United States Board of Supervising Inspectors of Steam Boilers requires 1.5 times the steam pressure on ordinary boilers, it requires that the hydrostatic pressure be twice the steam pressure on all coil and pipe boilers. The American Boiler Manufacturers' Association recommends that the hydrostatic test shall not exceed the working pressure by more than one-third of itself, and this excess is limited to 100 pounds. The rules formulated by the Board of Boiler Rules of the Commonwealth of Massachusetts make the hydrostatic pressure equal to 1.5 times the maximum allowable working pressure, though twice the maximum working pressure may be applied to boilers to carry not over 25 pounds, or on pipe boilers. The British Board of Trade and Lloyds require twice the working pressure; the National Boiler Insurance Company of England 1¾ times the working pressure, and the Scottish Boiler Insurance Company, of Scotland, for pressures above 80 and under 150 pounds —the working pressure plus 80 pounds; above 150 pounds—the working pressure, plus one-half the working pressure, etc., all showing that there is little uniformity in regard to this ratio.

We do not wish to be understood as condemning the hydrostatic test; it is an excellent thing in its way, and at times very valuable. In the case of small boilers, which cannot be entered for inspection, or in specially fitted work, or in testing cast-iron boilers, or in special vessels such as jacketed kettles, whose interiors can be viewed only through a 2-inch plug hole, the hydrostatic test is practically the only available means of testing. In such cases, when that test has been applied, all that it is possible to do has been done. The hydrostatic test is also valuable in testing new boilers in the boiler shop, to determine whether or not they are tight.

In a large measure, it is unfortunate that the hydrostatic test does not give definite and absolute quantities, for if this were the case, boiler inspection, which is now a rather disagreeable job, would be made quite easy and pleasant, and instead of crawling through small holes into hot and dirty interiors, and later making many computations on the strength of each part of a boiler, the

efficiency of the riveted seams, strength of the braces, resistance of the flat places, etc., all an inspector would have to do would be to keep his eye on the gage.

Since there is no connection between the hydrostatic and the allowable steam pressure, the latter can be stipulated only after an inspector has carefully examined the boiler inside and outside, taken careful measurements, computed the strength of the weakest part, and applied to that part a factor of safety approved by the dictates of good engineering practice.

Depreciation

In estimating the cost of power, or in making an appraisal of a plant, no one item is as uncertain and as unsatisfactory as that of depreciation. Interest can easily be reckoned as that which the same capital would return if safely invested in a bank or in securities; taxes and insurance may readily be determined on the basis of local conditions; but when it comes to reckoning maintenance and depreciation, doctors disagree.

By maintenance, we mean the cost of keeping the machinery and equipment in fair working condition, irrespective of its "age, color or previous condition of servitude." This can only be determined in each type and size of plant by keeping careful records. In large central stations, the total cost of repairs on buildings and on steam and electric equipment is usually about one-eighth of the total expense of running the station.

If proper attention be not given to maintenance, depreciation increases alarmingly, but the most careful repairing will not prevent the gradual reduction of values.

Depreciation is of two kinds, direct and indirect, or perhaps it were better to say, normal and accidental. Normal depreciation represents the gradual decrease of value and of efficiency due to age and wear.

The life of an engine or boiler is generally assumed to be from twenty to twenty-five years and the same rule holds for the greater part of the auxiliary equipment. Coal and ash-handling machinery and mechanical stokers probably have a shorter term on account of their continual motion and the severe service to which they are subjected, while the natural life of a condenser or pump may be considerably greater than that of the engine. These facts would point to five per cent. a year on the initial cost as a safe amount to allow for depreciation. This may be considered the normal depreciation of a power plant. If, however, we examine the history of most installations of this character, we will find that the term of life has been much shorter than that above mentioned and that this reduction in term of service is not due to wear or corrosion but to the fact that the plant has become old fashioned or that it has insufficient capacity. The increasing demands for power in a prosperous business may justify a radical policy in discarding the old plant entirely and substituting for it new machinery in larger units and more economical in operation.

While a fifty- or seventy-five-horsepower slide-valve engine and a fire-tube boiler carrying one hundred pounds pressure may be the most economical arrangement for a small plant, an increase to two hundred of three hundred horsepower would make it desirable to put in a four-valve engine and sectional boilers capable of carrying higher pressures. Although the discarded machinery may have some value, it is not such as to discount on very much beforehand and it may mean a practical scrapping of the whole plant in ten years instead of twenty.

Aside from this question of growth, there is the other one of improved machinery. A man who installs an engine and generator today is not at all sure but that in ten years there will be such improvement in both the steam and electrical ends of the machine as to make it desirable to change. This is particularly true in central power stations where large engines, steam turbines and generators are employed. There have been a number of instances within the past ten years where units which, when installed, were the best of their kind, have been condemned and replaced by others within seven or eight years, the enormous expense being justified by the fact that the amount of steam and coal used per kilowatt-hour was reduced about one-half by the change.

In these days of close figuring and keen competition, it is not possible for a company to do business with old-fashioned, uneconomical machinery. This problem of indirect depreciation has confronted manufacturers of machinery in much the same way. The invention of the Taylor-White process of making tool steel some ten years ago and the subsequent developments in the manufacture of various alloys has revolutionized machine-shop practice. A lathe which formerly plodded along contentedly, driven by a two-inch belt and consuming perhaps one-half a horsepower, has been replaced by a motor-driven machine using ten times that amount of power. It is not at all uncommon today to see machines using from ten to twenty horsepower in cutting cast iron and steel with high-speed tools. The lathe, planer or milling machine which is in good repair and practically just as capable of doing its intended work as when new, has necessarily been discarded and replaced by a much higher-priced machine.

This same cause has produced another revolution in the boiler and engine room. The seventy-five-horsepower engine and boiler are no longer able to supply the new demand for power and have been replaced by a plant of much larger capacity. It is true that this has all been a profitable investment and is now paying a good interest, but, nevertheless, it has involved a practical abandonment of the old machinery and a sudden increase in the capital invested. Examples like this show why manufacturers and owners of power plants find it necessary to use a large depreciation factor in estimating the cost of power or the overhead charges in the machine shop. A depreciation factor of from eight to twelve per cent. is not at all unusual nor excessive. Of course, no hard and fast rule can be laid down and each owner will have to decide this question for himself; but it is apparent that in this age of close figuring, no owner of machinery can afford to neglect a careful study of this problem. Either by the maintenance of a sinking fund or close attention to immediate profits, he must make provision for these sudden changes which are liable to be necessary.

Some business men prefer to estimate depreciation as a gradually decreasing factor rather than as a definite per cent. each year. For example, if five per cent. of the original cost is assessed each year, this would mean an estimated life of twenty years for the plant. Now, it is a fact that the depreciation, as a mere question of market value, is more rapid at first; that is, it would hardly be possible to sell an engine or a boiler at the end of one year for ninety-five per cent. of its face value or at the end of two years for ninety per cent. If ten per cent. depreciation were allowed each year on the inventory value of the preceding year, instead of a straight-line depreciation of five per cent., we should have a curve which falls at first rapidly and then becomes nearly parallel to its axis. By such a method, the inventory value at the end of ten years instead of being fifty per cent. would be about thirty-five, while at the end of twenty years instead of none, we would have a value of about twelve per cent. Such a system gives probably a more correct definition of values from year to year and would be preferable in determining inventory values. As a matter of determining the cost of power or the overhead charges in the shop, the other method is probably just as reliable.

Coal which costs the least per ton often proves to be the most expensive when results are considered.

If everyone knew when to put up, pay up and shut up, how few arguments there would be.

When an engine joins the "Knockers" club, there is a reason; when an engineer does so, he seldom has any excuse.

Chimneys and Draft

By John S. Leese

Temperature at which maximum draft is secured. Formula for estimating the draft which a given chimney is capable of developing. Types of chimneys and materials of construction.

Natural Draft

The natural draft obtainable from a chimney depends upon the column of gas inside the chimney being lighter than the air outside. The fuel consumption of a furnace using natural draft is not generally more than about 20 pounds of coal per square foot of grate area per hour, but with forced draft this consumption may be increased fourfold with equally complete combustion. If a pound of coal requires 12 pounds of air for perfect combustion, the natural-draft furnace will take about 24 pounds in practice, the difference being dilution. With forced draft less diluting air is required, because of the intensification and localization of the process of combustion; then 18 pounds or less of air will suffice. Therefore, as less air is required for forced draft than for natural draft to burn the same amount of coal, the furnace temperature will be higher, the heating surfaces will be more effective and the heat lost by hot gases is reduced, providing that the heating surface is large enough to allow them to leave the boiler flues at the same temperature as they would with natural draft. When forced draft is employed there is no necessity to allow them to leave the flues at a high temperature; the draft is there already, so that the escaping gases need only be reasonably hotter than the feed-water temperature. With natural draft this heat is wanted to make the draft. In practice these theoretical advantages of forced draft are not always available as forced-draft systems are generally installed to "force" the boilers.

The most efficient boiler installation would be one with a heavy forced draft, a proportionately small grate area and a proportionately large heating surface, the latter being further developed by the use of an economizer. The waste gases would then leave the economizer as cool as possible and the important air and coal factor would be as small as practicable for good combustion. If

T = absolute temperature of the gases inside the chimney in degrees Fahrenheit, and

t = absolute temperature of the outside atmosphere in degrees Fahrenheit, then, to get the best natural draft, the approximate relation is

$$T = 2.1\ t.$$

Taking an average of 520 degrees Fahrenheit for t, then

$$T = 2.1 \times 520 = 1092$$

degrees Fahrenheit absolute, or = 630 degrees above Fahrenheit zero.

Any temperature above or below this will be less efficient in inducing air. At this temperature a stage is reached where the gain in head and the strength of the current due to it are nullified by the diminution of density and the draft is actually reduced. It should, therefore, be remembered that there is no advantage whatever in carrying a higher waste-gas temperature than about 600 degrees Fahrenheit, and, that from the point of view of thermal efficiency, the lower the chimney temperature, the better because less heat will go to waste.

With natural draft and its most economical chimney temperature as obtained above, about one-fifth of the total energy contained in the coal will go up the chimney. If the temperature of the products of combustion leaving the boiler is more than 600 degrees Fahrenheit, it is, then, desirable to use an economizer.

The draft (d) in inches of water can be approximately obtained by the formula:

$$d = H\left(\frac{7.6}{t} - \frac{7.9}{T}\right)$$

where

H = hight of chimney above grates,

T and t are the absolute temperature as before.

If the above average values are given to T and t, we get

$$d = H\left(\frac{7.6}{520} - \frac{7.9}{1092}\right) = 0.0067$$

or

$d = 0.007\ H$ inches of water, approximately

and

$d_1 = 0.2\ H$ feet per second, approximately.

When

H = Hight of chimney above fire bars in feet,

C = Pounds of coal fired per square foot of grate area per hour,

a = Grate area in square feet,

E = Effective area of chimney in square feet,

then

$$H = \frac{a^2\ C^2}{200\ E^2} = \frac{(Total\ coal\ per\ hour)^2}{200\ E^2}$$

and hence:

$$E = 0.08\ \frac{a\ C}{\sqrt{H}}$$

If A equals actual area of chimney in square feet,

$$A = E + 0.7\sqrt{E}$$

and

$$E = A - 0.6\sqrt{A}$$

The capacity of a chimney varies as the square root of the hight and directly as the area.

If P equals the actual horsepower required in use, then

$$P = 3\tfrac{1}{3}\ E\sqrt{H}$$

$$H = \left(\frac{0.3\ P}{E}\right)^2$$

$$E = \frac{0.3\ P}{\sqrt{H}}$$

If a forced draft is obtained by means of a steam blast, as used on locomotives, the draft in inches of water is very approximately obtained with short chimneys by Longridge's formula:

$$d = \frac{37\ D^{1.662}\ p^{0.8}}{S^2}$$

where

d = draft in inches of water,

D = diameter of jet or blast pipe in inches,

p = pressure of steam in pounds per square inch,

S = diameter of chimney in inches.

With natural draft and low-grade fuel the hight of a chimney is seldom less than 70 feet, and the ordinary hight of a mill chimney is about 200 feet. If more boilers are being added to a battery already installed and connected to the same chimney the hight should be increased in a greater ratio than the area.

Types of Chimneys

The wind pressure in designing chimneys may be calculated at 56 pounds per square inch. If unity is taken for a square chimney the relative pressure on a hexagonal chimney will be 0.75, for an octagonal chimney 0.65 and for a round chimney 0.5. Hence, a round chimney is preferable to a square one in outside section, because it has less windage, and as regards inside section, because the friction and eddy-current effects are less.

The cheapest chimney is the one built up of steel plates, riveted together and unlined with brick. These should be suitably anchored by guy-stays. The self-supporting steel chimney is bolted to a foundation which is massive enough to withstand windage stresses. The durability of steel chimneys depends on the thickness of the plates, the kind of fuel used, the atmospheric conditions, the condition and state of maintenance. Steel chimneys should be painted frequently

with the best quality of paint in order to stop and prevent corrosion. Next in order of cost comes the steel chimney partly lined with brick. Hard selected-brick chimneys, double-walled, are the next in cost. These should have an internal core such that it is free to expand without straining the outer supporting core. "Custodis" chimneys are becoming very popular. They are built without a core of specially molded perforated bricks and are more costly to erect than other types, but it is claimed that the maintenance is very low. Good bricks should give a metallic "clink" when knocked together. The absorption of water by bricks should not be more than one-sixth their weight, and no bricks with cracks, flaws or lumps, or which are not true in shape, should be used in chimney construction. Some facing bricks will harden on exposure. Brickwork weighs about 100 to 130 pounds per cubic foot. The total weight of a chimney may be approximately arrived at by the following:

For square chimneys, *Weight* = 56 × *breadth of base* × *area exposed.*

For octagonal chimneys, *Weight* = 35 × *breadth of base* × *area exposed.*

For round chimneys, *Weight* = 28 × *breadth of base* × *area exposed.*

The width of the base of a square chimney should be at least one-tenth the hight and for other shapes not less than one-twelfth. The batter should in no case be less than 0.25 inch per foot of hight. The thickness of the brickwork at the top of a chimney should be at least 8½ inches, or one brick. This thickness may extend 25 feet down. From 25 feet from the top to 50 feet from the top the thickness should be 1½ bricks, increasing ½ brick every 25 feet down. If the inside diameter of the top of the chimney is more than 4 feet 6 inches the top length should be at least 1½ bricks thick and so on.

The construction of chimney foundations is most important. After putting in the base, the concrete and brickwork must be allowed to settle before the stack is built. All brick chimneys should have cast-iron caps, well clamped to the brickwork. Copings should not be heavy or large to catch the wind and must be bound securely together. For this purpose copper cramps are preferable to slate dowels. The center of gravity of the coping in section must fall within the outer line of the shaft. In conclusion, all chimneys should be fitted with a substantial lightning conductor.

Questions for Hoisting Engineers

Following is a list of printed questions used at an examination for hoisting engineers held at Terre Haute, Ind., May 10 and 11:

Answers are not particularly desired to these questions, but a discussion of any or all of them as to their fitness for the purpose and their completeness will be welcome.

It should be understood that an examination is not held for the purpose of asking questions which the candidate cannot answer, but to find those which he can. He is expected to demonstrate his fitness to care for and operate the machinery to be placed in his hands, and this may be as clearly shown by intelligent answers to simple questions as by halting solutions which just "get by" the more complex ones.

1. Name the different capacities in which you have been employed in or about coal mines, giving the length of time you were employed in each capacity and the different localities where you have been employed.

2. (a) What are the duties of a hoisting engineer, as required by law? (b) What duties are incumbent on an engineer other than those required by law?

3. Steam boilers, water-fed. (a) Explain how it is possible to pump water into a boiler by means of its own steam pressure. (b) What prevents the water from running back after the stroke of the feed pump is finished?

4. (a) What is the best rule for finding how much pressure a boiler will actually stand? (b) What is the factor of safety usually allowed between working pressure and bursting strength?

5. (a) A return-tubular boiler is 72 inches by 18 feet long. What is the bursting pressure? (b) What would be a safe pressure? (c) What factor of safety is used when new?

These questions were used in an examination of applicants for certificates of competency to serve as hoisting engineers in the State of Indiana and are published to give the reader an idea of the attainments required.

6. (a) A pair of hoisting engines have cylinders thirty inches in diameter and sixty-inch stroke. (b) With a steam pressure of ninety pounds per square inch, how many revolutions per minute must the engine make to develop 1000 horsepower?

7. What is lap? What is lead? What is mean effective pressure?

8. (a) What is the cause of water foaming? (b) How can foaming in a boiler be detected?

9. How should the slide valve and pistons of an engine be tested to determine whether they leak?

10. What may be done to remedy the following faults: (a) A too early cutoff. (b) A too late cutoff.

11. State fully the various points to be considered in the arrangement of the tubes in a return-tubular boiler.

12. A weight of 5000 pounds is lifted by a hoisting engine to a hight of 400 feet. What is the theoretical quantity of heat consumed in the operation?

13. What three elements enter into each calculation relating to the safety valve?

14. A boiler 48 inches in diameter is subjected to a pressure of 100 pounds per square inch.

If the boiler plate is three-eighths of an inch thick, what is the stress per square inch of section along the sides?

15. What different kinds of stress or strain are placed upon the different materials used in a hoisting outfit, including the boilers?

16. (a) What is the best way to start a jet to draw water 185 feet vertically? (b) Explain how a duplex steam pump works.

17. If the forward eccentric rod were to break, could the backward eccentric rod be used to run the engine forward? If so, how?

18. (a) Does the acting eccentric lead or follow the crank when the link is in full gear? (b) In what position would you place the link motion when the slide valve is to be set? (c) What determines the length of the eccentric rod when setting the slide valve?

19. (a) Why should the temperature of the boiler furnace be kept as high as possible? (b) What causes the reduction of this temperature? (c) What care must be exercised in burning bituminous coals containing hydrocarbons?

20. (a) Name and define the principal parts of the machinery used in and around coal mines in this State.

21. How should a pump or an injector be worked when feeding more than one boiler?

22. (a) What effect does increasing the lead have on various operations of the valve? (b) When the diameter of a pinion or gear is known, also the pitch of the teeth, what is the rule for determining the number of teeth?

23. (a) State the three relations between force and motion. (b) What is the allowable working load for a steel wire rope 5¼ inches in circumference.

24. Why is a cylinder counterbored?

25. Show clearly the difference between a throttling governor and the automatic governor.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it.

The First Locomotive

My friend claims that Stephenson invented the locomotive. I claim that a man named Murdoch built a model before Stephenson did. Please decide the matter.

R. W.

The idea of applying steam to locomotion was probably first suggested by Robinson, in 1759. In 1784 Watt patented a locomotive; but he did not build it. About the same time Murdoch, who was Watt's assistant, made an efficient working model. Trevithick and Vivian built a locomotive about the year 1804, which traveled at five miles an hour and carried a net load of about ten tons. Blackett and Hedley discovered, in 1813, that the necessary tractive power could be obtained without the use of rack and pinion. Stephenson's "Rocket" was not built until about 1829; but it had a blast pipe and a tubular boiler, so that it may be fairly regarded as the parent of the lomocotive of today. See Rankine's "Steam Engine."

Gas Engine Temperatures

Is it true that the gases in a gas-engine cylinder are not at the same temperature throughout, and if so, why?

K. L. D.

It is, especially in large cylinders; because gases are poor heat conductors, and the heat in the gases which are in the center of the cylinder cannot get to the walls as fast as heat is taken from the gases there by the jacket water.

Flame Propagation

What is meant by "flame propagation?"

S. N. C.

The spread of flame through combustible gases. If a combustible mixture of gas and air were inclosed in a tank and ignited at one point, the flame would be communicated from the burning gases to those immediately next to them; when these kindled, the flame would spread to the next unlighted gases, and so on throughout the whole tankful of mixture. That is flame propagation. It occurs so rapidly that it seems instantaneous.

Sensible and Latent Heat

What is the difference between sensible and latent heat?

J. M. C.

Sensible heat is that which may be measured by the thermometer. Latent heat does not affect the thermometer. It is the heat that is absorbed by a body when passing from a solid to a liquid or from a liquid to a gaseous state without change of temperature.

Pump Discharge

A pump sends water through 1500 feet of 2-inch pipe, and then raises it into a tank, the water in which is 24 feet higher than the feed pipe. There are two inlets to the tank, one at the bottom and one rising over the side of the tank and passing in at the top. Which of the discharges will be easier on the pump?

N. W.

If the upper pipe dips into the water in the tank, there will be no difference. If it does not dip into the water in the tank, but empties a short distance above it, then the bottom discharge will be the easier.

Pitch of Propellers

What is the pitch of a screw propeller and how is it found?

J. R.

The pitch of a propeller is the degree of spirality or the distance it would advance in one revolution if the water in which it turned was solid and there was no slip. It is measured by drawing a line representing the shaft and at right angles to it another in its length representing the circumference of the circle made by the tips of the blades. Through the intersection of these lines a diagonal line is drawn at an angle equal to that which the outer edge of the blade makes with the shaft. The perpendicular distance from the end of the line representing the circumference to the diagonal line is the pitch in the scale to which the lines are drawn.

Bubbling Transformers

Two oil-cooled transformers hum excessively and throw oil out of the cases when the load is greatest. How can this be remedied?

B. R. P.

Try reducing the load on the transformers. Both the loud humming and oil bubbling are due to overheating. This may be caused by overloading, or the windings may be "leaky," permitting excessive local currents to flow, but the latter is not probable.

Chimney Dimensions

Will you tell me how to get the hight of a chimney without using a transit?

C. E.

It may be determined by comparing the length of its shadow with the length of the shadow of any object whose hight is known. Suppose the length of the shadow of the chimney to be 94 feet and the shadow cast by a board 12 feet long standing vertically is 8 feet long, the hight of the chimney will be to the hight of the board as the length of the chimney shadow is to the length of the shadow of the board, or 141 feet. Stated in the form of a proportion the solution reads

Lap and Lead

Is it necessary to have lap and lead on a slide-valve engine?

S. R. B.

That depends on the definition given to the word necessary. Lap is given to a valve to enable steam to be cut off before the end of the piston stroke and allow expansion to take place in the cylinder and thus reduce the quantity needed for the work. Lap reduces steam consumption within certain limits and is necessary if economy is desired.

Lead provides for full steam-chest pressure on the piston at the beginning of the stroke and is considered necessary by the majority of engineers.

Low Voltage and Hot Field Windings

A 2200-volt alternator will not keep its voltage quite up to normal at full load, although the field is so heavily excited that the magnet coils run hot. The motors on the circuit run too slow during the peak load. What is the trouble?

B. R. P.

The machine is not up to its rated speed; this causes the drop in motor speed, and helps to pull the voltage down; the low power factor of your load also pulls down the voltage, in spite of the excessive current delivered to the field winding by the exciter. Your engine regulation is also probably poor.

Strain on Staybolts

Please work an example for me by the rule for finding the strain on staybolts, given in POWER for February. Suppose that the bolts are four inches apart, that the gage pressure is 140 pounds.

C. R.

Following the rule, $4 \times 4 = 16$, and $16 \times 140 = 2240$ pounds. The stress is therefore 2240 pounds, or a little over a ton.

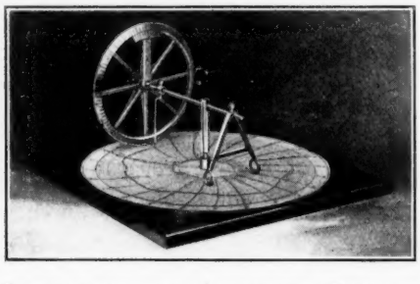
New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news.

An Averaging Instrument for Circular Charts

The Bristol-Durand radii averaging instrument for circular-chart records from recording meters is shown in the accompanying illustration. This instrument was devised by Prof. W. F. Durand, of the Stanford University, and is manufactured by the Bristol Company, Waterbury, Conn. The instrument is used for determining the average reading for a given circular-chart record, such as is drawn by recording pyrometers, thermometers, pressure gages, voltmeters and similar instruments.

The simple construction of the instrument is shown by the illustration. A wooden base with a metal socket is provided for supporting and centering the chart. The socket holds a rotatable pin which has a vertical slot at the top to receive the bar which carries the tracer point and triangular support. The vertical groove in the pin allows the integrating wheel to roll on the chart with uniform pressure due to its own weight.



INSTRUMENT FOR AVERAGING CIRCULAR CHART RECORDS

The integrating wheel is six inches in diameter; the rim is graduated into one hundred equal spaces, and is fitted with a vernier, which makes it possible to read easily with the naked eye to one-tenth of one division on the wheel. The wheel is of such large size it is not necessary to supply any counting device for the number of revolutions. The number of complete revolutions cannot be more than two, even for a record of maximum size on the large 12-inch charts.

To operate the instrument the thumb and forefinger of one hand are applied to the base of the triangular support, which is moved radially, so as to cause the tracer point continually to follow the record curve; the chart is turned with the other hand.

The principle upon which the device depends is that sliding of the horizontal shaft back and forth in the slot of the central post does not produce any rotation of the wheel. The amount of rotation depends upon the length of all the elementary circular arcs of which the irregular curve may be considered to be composed. As the length of a circular arc within a given angle is in direct proportion to its radius, the measurement of the sum of all these elementary arcs is in direct proportion to the average radius, which is the dimension desired. The instrument is equally applicable to charts having either straight radii, or circular-arc radii.

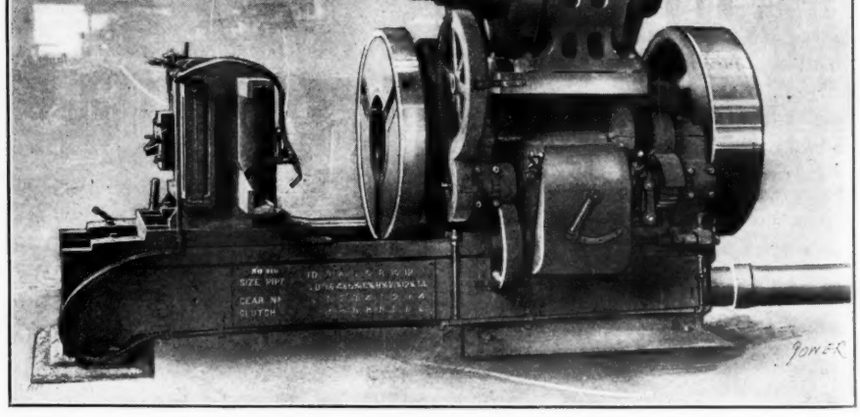
New Motor-Driven Pipe-Threading Machine

The motor-driven pipe-threading machine shown by the accompanying picture is one of several recently installed by the Stoever Foundry and Manufacturing Company, of Myerstown, Penn., at the pipe mill of the Spang-Chalfant Company, Sharpsburg, Penn. The capacity of the machine is from 4 to 12 inches inclusive. The installation is of especial interest in that this threader is the fastest machine for threading or cutting pipe on the market. It will thread pipe at the rate of 28 feet per minute, a very fast speed for a tool of this size.

The machine is driven by a Westinghouse squirrel-cage induction mill motor, which has the rugged construction and the general reliability required for such service. The normal rating of the motor is 10 horsepower, and the full-load speed is 875 revolutions per minute on a three-phase 25-cycle 200-volt circuit.

The pipe-threading machine has ten speeds; the gear box gives five changes, and a double train of gears outside of the box give two speeds for each one in the box. All of the gears are made of carbon-steel castings and are fitted with bronze bushings. The gears that are subjected to the most wear have case-hardened teeth. All the gears run in oil. A number of changes in speed can be made without stopping the machine. The machine can be started or stopped by means of a clutch on the operating side of the tool. The gearing is on the outside of the machine and accessible to the operator. An internal driving gear placed in close to the body of the machine is used on the rear of the machine.

The bed of the machine is so designed and constructed as to give a very stiff and strong cross-section. The bed is supported on the outer ends by suitable legs, and under the headstock by a very heavy pedestal base that strongly supports the entire driving mechanism of the machine. The headstock of the machine is made in one casting, similar to that of a lathe; this one casting contains all the principal bearings, which insures permanent alinement and great rigidity. The spindle or



MODEL H MILL TYPE PIPE-THREADING MACHINE

arbor of the machine is of such a diameter on the inside that the largest-sized fittings within the capacity of the machine can be passed through.

There are provided two gripping chucks of unusually heavy design, made in one piece without any separate parts or faceplates. Each chuck has three independent jaws operated by powerful screws. These jaws are made of steel and are equipped with hardened-steel plates for holding the pipe.

The die head is of the sliding or floating type that allows for any eccentricity in the pipe. The die head can be pushed to one side and the pipe cut off at a distance of 3 inches from the face of the front gripping chuck. This type of head permits inserting the pipe from the front or rear of the machine without injury to the dies. Rounding blocks are provided for rounding the pipe before it is threaded.

The adjusting mechanism is maintained in a straight line to prevent the digging of the chasers into the pipe. This mechanism consists of but two principal parts: a lever for opening and closing the dies, and a screw for making the adjustments to gage. The chasers are single pieces of steel without any pins or links. They are made by a special process so that any single chaser in a set can be replaced without reference to the rest of the set.

The oil pump on the machine is of the geared type and is driven direct from the constant-speed shaft. Valves are placed at both the die head and the cutting-off tool to regulate the supply of oil at these points. All surplus oil flows back into the reservoir in the bed of the machine through a relief valve.

All of the bearings of the pipe threader are provided with long and deep receptacles for holding lubricating wool and oil so that the machine is operated at all times with perfect lubrication.

Lisk's Universal Saddle

These saddles are made of cast iron, in various sizes, for supporting tanks, feed-water heaters, internally fired boilers, etc. They are light in weight, and can be used for different tank diameters by changing the length of the crossbars. The bearing pads are adjustable, so that they will conform to the curvature of the tank body.

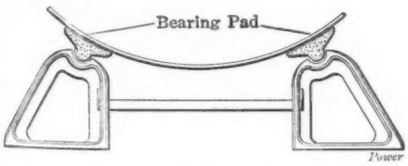


FIG. 1. TYPE A SADDLE AND FRAME

Fig. 1 shows the design of what is known as "Type A," which consists of a set of saddles made up of four pads and frames, four bases and two crossbars. It is used as shown in Fig. 3.

Fig. 2 shows the "Type B." This type of saddle is for use when it is desired to elevate the tank, heater, etc., some distance above the floor.

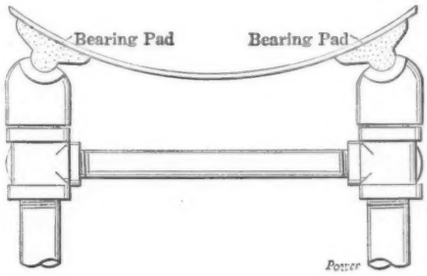


FIG. 2. TYPE B SADDLE AND FRAME

These saddles are made by J. P. Lisk, 151 Quincy street, Brooklyn, N. Y.

The Teller Air Burning Furnace

This furnace is designed to accomplish efficient and smokeless burning of fuels containing high percentages of volatile matter, such as are found in the Middle West. It is constructed on the Dutch-oven principle. The illustration shows it applied to a horizontal return-tubular boiler, but it is also being used in connection with Scotch and water-tube types of boiler.

The furnace proper consists of a double arch of specially designed blocks of steel mixture, reinforced on the outside with a heavy wall of ordinary brick and a supplementary arch above, through which air ducts are provided for admitting air to the fire, as shown. The air ducts are placed at a point that will permit the air to mingle with the products of distillation just as they are passing from the furnace, enabling them to thoroughly mix and burn, before coming in contact with the heating surface.

The inverted firebrick bridgewall is a characteristic of the design and assists in mixing the products of combustion. An air damper is arranged, as shown at *A*, at a convenient point for the firemen to manipulate. Provision is made in the front wall of the furnace for observing the condition of the fire, so that the air may be regulated accordingly.

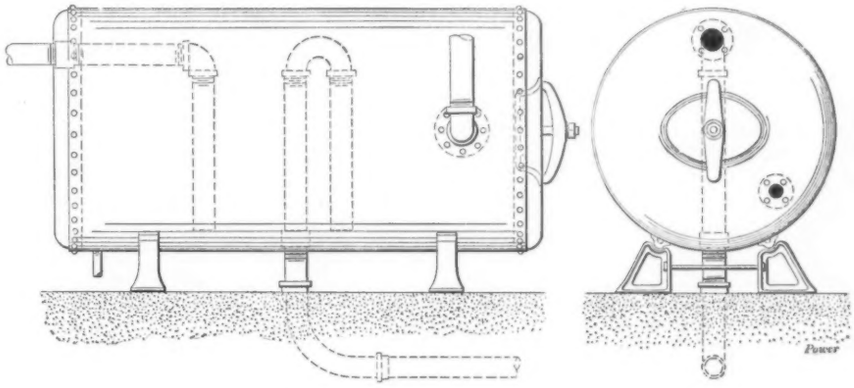
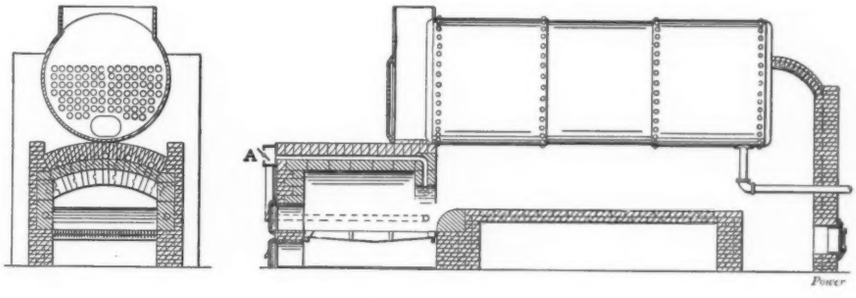


FIG. 3. TANK RESTING ON SADDLES AND FRAME

The bridgewall proper is 12 inches high and the grates are set 48 inches lower than the shell of the boiler. The bridgewall is continued back to a point just below the blowoff, the idea being to confine the gases within a reasonable distance of the heating surface. This furnace is regularly equipped with an efficient rocking grate. It is manufactured by the Teller Air-Burning Furnace Company, Rogers, hotel, Minneapolis, Minn.



TELLER'S AIR-BURNING FURNACE

In the period 1846 to 1855 only 8,000,000 tons of coal were produced in this country. The tonnage increased thence up to the period of 1876 to 1885, when it was 84,000,000 tons. During the period 1896 to 1905 it increased to 283,000,000 tons, and in the last six years it increased to 436,000,000 tons.

Engine Wreck at Stafford Mill

On Tuesday morning, June 7, as the engine in No. 1 Stafford mill, Fall River, Mass., was being started, the vacuum was lost and as one of the attendants opened the forced-injection valve while the other closed the throttle through some misunderstanding, water found its way into the low-pressure cylinder, causing a serious wreck. As the water was caught in the head end of the cylinder the shaft was forced out of the frame, and the rim of the flywheel striking the side of the foundation broke one of the segments. A detailed description of the accident will appear in a later issue.

Locomotive Boiler Explosions

Blanks were sent by a committee appointed by the American Railway Master Mechanics' Association, upon Design, Construction and Inspection of Locomotive Boilers, to all of the principal railroads in the United States, asking for information in regard to boiler-inspection rules and regulations, and also as to casualties resulting from boiler explosions of all kinds. Replies were received from 157 railroads, giving the number of boiler explosions and failures, and casualties to employees and others, resulting therefrom during the period from January 1, 1905, to November 1, 1909. These 157 railways own and operate 43,787 locomotives and 157,169 miles of roadway, and during the period above mentioned made 6,012,057,467 locomotive miles. The committee estimates that the reports cover over 75 per cent. of the locomotives in use in the United States.

Explosions and failures of locomotive boilers are divided into five classes, as follows:

Explosions of boiler shells.
Explosions of fireboxes.
Damage by burning.
Rupture of flues.
Boiler-pitting failures.

Of the first three classes 98.3 per cent. were due to low water, and 1.7 per cent. to other causes. Of the failures due to low water 98.6 per cent. were due to the failure of men handling or in immediate charge of the locomotives to maintain a proper supply of water in the boilers. The remaining 1.4 per cent. were due to other causes.

Of the 407 killed and injured, 386, or 94.8 per cent., were due to accidents caused by low water, while the remaining 21, or 5.2 per cent., were from other causes, some of these being the result of or incident to wrecks, and a small number are thought to be due to accidents caused by defective design, material, workmanship, or to the poor condition of the boiler or fittings; but it is doubtful if any of them could have been prevented by any method of inspection, in addition to that which is now in force. The other classes of failures were as follows:

While the number of flue ruptures appears in itself to be large, it covers the record of an average number of 42,200 locomotives per annum, for a period of four years and ten months. Allowing 250 flues to each boiler, the result shows one flue failure per year to each 15,912 flues in service, or a percentage of only 0.00006 of 1 per cent. on an average 1,876,422 miles were made for every tube that was ruptured.

From Senate document No. 682 it appears that the average number of employees killed and injured per annum on account of boiler explosions and locomotives for the period from August 1, 1903, to November 1, 1908, was 47.7 employees and others killed, and 134.2 injured. The Senate documents include all the locomotives in use during the time mentioned. The committee's returns show an average of 38 killed and 63.1 injured for about 75 per cent. of the locomotives.

Suppressing Smoke in Milwaukee

According to the records of the smoke inspector's department in Milwaukee, there are 478 steam plants in operation in the city, containing a total of 1080 boilers. On the whole the plants are reported as not giving a great deal of trouble in the way of smoke. Nearly every make of stoker is represented in the equipment of these plants as well as various modifications of hand-fired furnaces. Tugs in the river are still a great nuisance to the smoke department and locomotive smoke is also a factor. The Northwestern is reported as having the best system of smoke suppression, the road employing a smoke inspector who devotes his entire time to the Milwaukee yards, while the St. Paul road is not so satisfactory, as the inspector for this road is required to divide his time between Chicago and Milwaukee.

An interesting series of tests was recently made to determine the accuracy with which a camera would record smoke coming from a locomotive. Railroad men claimed that the vapor caused from the steam blowers would show as heavy smoke in the camera. It was demonstrated, however, that the camera was not at fault in this regard.

Formerly it was difficult for the smoke department to obtain convictions on the report of its observers, the steam users insisting that they had not smoked at the time specified. There has now been worked out, however, a system of charts accompanied by photographs which has proved entirely satisfactory to the department. These charts divide the time up into minutes, the observer making note of the character of the discharge from the chimney for each minute during the hour. Accompanying this chart is a series of photographs of the stack with the time marked on the chart and also on the photograph. Using this system the department has never failed to obtain a conviction where a steam user has violated the smoke ordinance.

In this regard the annual report of the smoke inspector (Charles Poethke, third floor, City Hall) for the year 1909, issued in bulletin form, will undoubtedly be of interest. The bulletin tells of the progress made in Milwaukee in preventing smoke and gives the city ordinances relating to the suppression of smoke.

NEW PUBLICATIONS

ELECTRICITY. By H. M. Hobart. Published by D. Van Nostrand Company, New York. Cloth; 225 pages, 5½x 8½ inches; 115 illustrations; 43 tables. Price, $3.

Like all of Mr. Hobart's writings, this book has a strong practical-engineering flavor and his treatment is not restricted to the well beaten track commonly followed by textbook writers. The features which incite criticism are very few; the most objectionable one, so far as American readers are concerned, is the use throughout the work of metric units and the Continental practice of using commas for decimal points. To a student unfamiliar with this latter custom, the tables and numerical examples are likely to be confusing if not actually unintelligible.

A second fault is the use of arbitrary terms as though sanctioned by common usage or recognized authority; for example, one kilowatt-hour is called a "kelvin" and one watt-hour a "siemens." Both are universally honored names but their application to these units has never been approved by any authoritative body.

The author's use of the ton as the practical unit of weight is also unfortunate. Few students are accustomed to think in any such huge units; moreover, as the author is unwilling to talk in English units it would have been wiser to avoid the use of a term that might be mistaken for one.

In other respects, and especially as to technical accuracy and clarity of exposition, the book is excellent. It is not so much an elementary textbook as it is a presentation of fundamental principles. Chapter X, on Inductance, is particularly lucid and in Chapter XII, on Insulating Materials, the experience of the author in designing electrical machinery bears good fruit. We hope that an early future edition will be expanded to make a thoroughly comprehensive treatise; if the author will do that, correcting the few existing defects and maintaining the otherwise high standard of the present edition, the result will be what the small boy would call a "crack-a-jack."

NEW INVENTIONS

Printed copies of patents are frnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

GASOLENE ENGINE. Earl Vanderlip, Alto, Mich., assignor of one-half to Arthur O. Bickford, Alto, Mich. 957,278.

INTERNAL COMBUSTION ENGINE. Chas. W. Higgs, London, England. 957,336.

MULTISTAGE TURBINE. Francis Hodgkinson, Edgewood Park, Penn., assignor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 957,338.

FLUID PRESSURE MOTOR. Harris B. Holt, Pittsburg, Penn., assignor of one-half to Charles A. Conn, Pittsburg, Penn. 957,339.

CURRENT MOTOR. Thomas J. Steele, Citra, Fla. 957,388.

ROTARY COMPOUND STEAM ENGINE. Charles R. Hilty, Chicago, Ill. 957,508.

ROTARY INTERNAL COMBUSTION ENGINE. James S. Stewart, Diamondville, Wyoming. 957,631.

INTERNAL COMBUSTION ENGINE. Paul Daniel, New York, N. Y., assignor to Benjamin G. Paskus, New York, N. Y. 957,657.

PNEUMATIC MOTOR. Thomas Danquard, New York, N. Y., assignor by mesne assignments to Auto Pneumatic Action Co., New York, N. Y., a Corporation of New York. 957,751.

INTERNAL COMBUSTION ENGINE. Nathaniel Metz, Lansdale, Penn., assignor of one-half to William D. Heebner, Lansdale, Penn. 957,788.

ROTARY ENGINE. Albert E. Rasmussen, North Yakima, Wash. 957,799.

BOILERS, FURNACES AND GAS PRODUCERS

FEEDER FOR FURNACES. Louis P. Bolander, San Francisco, Cal. 957,184.

MECHANICAL STOKER FOR FURNACES. Frank C. Armstead, Pittsburg, Penn., assigaor to the Westinghouse Machine Company, a Corporation of Pennsylvania. 957,296.

FURNACE OF STEAM BOILERS AND THE LIKE. Walter G. Crosthwaite, Leeds, England, assignor by mesne assignments to the United Cigarette Machine Company, Ltd., London, England, a Corporation of Great Britain. 957,310.

SMOKE CONSUMER. Thomas Rees, Chicago, Ill. 957,377.

FURNACE ARCH CONSTRUCTION. Colin Campbell, Rutherford, N. J. 957,494.

OIL BURNER. Valdemar F. Lassoe, Brooklyn, N. Y. 957,614.

BURNER. Oriel M. Graves, Spokane, Wash. 957,857.

POWER PLANT AUXILIARIES AND APPLIANCES

SURFACE CONDENSER. Louis A. Alberger, Greenwich, Conn., assignor to Alberger Condenser Company, New York, N. Y., a Corporation of New York. 957,175.

VACUUM TRAP. Leslie P. Strong, Cleveland, Ohio, assignor to the Clarke Manufacturing Company, Cleveland, Ohio, a Corporation of Ohio. 957,269.

VALVE FOR GAS ENGINES. Robert White, Beaver Falls, Penn., assignor of one-half to E. S. Burns, Beaver Falls, Penn. 957,285.

PISTON VALVE. John T. Wilson, Jersey Shore, Penn. 957,286.

VAPOR TORCH. Thomas Yates, Sunnyhurst, Darwen, England. 957,292.

STOP AND CHECK VALVE. George C. Davis, and Walter A. Daley, Chicago, Ill., assignors to George M. Davis Regulator Company, Chicago, Ill., a Corporation of Illinois. 957,311.

VALVE. Clifford A. Ellis, Bayonne, N. J., assignor to Standard Oil Company, of New York, a Corporation of New York. 957,318.

COMBINED STUFFING BOX, GASKET, VALVE AND SPRING. Guy L. Kennedy, San Francisco, Cal., assignor to National Carbonated Liquid Co., San Francisco, Cal., a Corporation of California. 957,347.

VALVE. Ernest E. Sweet, Detroit, Mich., assignor to Cadillac Motor Car Company, Detroit, Mich., a Corporation of Michigan. 957,391.

VALVE FOR STEAM TRAPS AND THE LIKE. Charles J. Mackerey, Newport News, Va. 957,469.

VALVE FOR STEAM TRAPS. Charles J. Mackerey, United States Navy. 957,470.

PRESSURE GAGE. Coleman J. Manning, Medford, Mass., assignor to American Steam Gauge and Manufacturing Company, Boston, Mass., a Corporation of New Jersey. 957,471.

DISTRIBUTING AND REVERSING EXPANSION VALVE. William M. Coward, Sydney, New South Wales, Australia. 957,499.

GAS-ENGINE PRIMER. Lawrence R. Darymple, Abilene, Kan. 957,500.

PRESSURE CONTROLLING VALVE. Frank A. Lockwood, Denver, Colo., assignor to the Denver Engineering Works Co., a Corporation of Colorado. 957,511.

VALVE CHAMBER FOR WATER COLUMNS. Frank C. Anderson, Cincinnati, Ohio, assignor to American Valve and Meter Company, Cincinnati, Ohio, a Corporation of West Virginia. 957,533.

VALVE. Everett P. Allen, Chicago, Ill., assignor to Norwall Manufacturing Company, Chicago, Ill. 957,592.

OIL-PROOF SPARK PLUG. Eli J. Bushey, New York, N. Y. 957,653.

CENTRIFUGAL PUMP. Richard O. Jones, Dayton, Ohio, assignor by mesne assignments to the Computing Scale Company, Dayton, Ohio, a Corporation of Ohio. 957,682.

COUPLING. Ernest Marek, Chicago, Ill. 957,695.

VALVE FOR INTERNAL COMBUSTION ENGINES. Francis W. Brady, New York, N. Y. 957,730.

CARBURETER. Francis W. Brady, New York, N. Y. 957,731.

REGULATOR FOR PNEUMATIC MOTORS. Thomas Danquard, New York, N. Y., assignor by mesne assignments to Auto Pneumatic Action Co., New York, N. Y., a Corporation of New York. 957,752.

SLIDE-VALVE MOTION FOR STEAM ENGINES. Eugen Frikart, Mulhausen, Germany, assignor to Elsassiche Maschinenbau-Geselischaft, Mulhausen, Germany. 957,762.

PUMP. Atwell A. Parker, Waterford, N. Y. 957,794.

LUBRICATION OF THRUST BEARINGS. Charles A. Parsons, Newcastle-upon-Tyne, and John Turnbull, Wallsend, England; said Turnbull, assignor to said Parsons. 957,797.

VALVE. Charles E. Strong, Amityville, N. Y. 957,811.

ELECTRIC INVENTIONS AND APPLICATIONS

ELECTRIC BATTERY. Frank A. Decker, Philadelphia, Penn., assignor to Decker Electrical Manufacturing Company, Wilmington, Del., a Corporation of Delaware. 957,192.

ELECTRIC GENERATOR AND MOTOR. Charles T. Hibbard, Minneapolis, Minn., assignor to Electric Machinery Company, Minneapolis, Minn., a Corporation of Minnesota. 957,209.

COIL FOR ELECTRICAL PURPOSES. Elbert W. Jodrey, Lynn, Mass., assignor to General Electric Company, a Corporation of New York. 957,213.

ELECTRIC HEATER. Judson C. Logan, Pittsfield, Mass., assignor to General Electric Company, a Corporation of New York. 957,226.

DYNAMO ELECTRIC MACHINE. Jakob E. Noeggerath, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 957,242.

DYNAMO ELECTRIC MACHINE. William D. Pomeroy, Norwood, Ohio, assignor to Allis-Chalmers Company, a Corporation of New Jersey, and the Bullock Electric Manufacturing Company, a Corporation of Ohio. 957,249.

ELECTRIC SOLDERING IRON. Edwin W. Rice, Jr., Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. 957,256.

DIRECT CURRENT TURBO GENERATOR. Edwin C. Wright, Newport, Ky., assignor to Allis-Chalmers Company, a Corporation of New Jersey, and the Bullock Electric Manufacturing Company, a Corporation of Ohio. 957,290.

BATTERY-CHARGING AND DISCHARGING SYSTEM. Lothar Fiedler, Stoke Newington, London, England. 957,321.

ELECTRIC FURNACE. Hiram W. Hixon, Philadelphia, Penn. 957,337.

ALTERNATING-CURRENT MOTOR. Valere A. Fynn, London, England. 957,505.

ELECTRICAL SYSTEM OF DISTRIBUTION. Walter E. Winship, New York, N. Y., assignor to Gould Storage Battery Company, a Corporation of New York. 957,530.

ELECTRIC ROSETTE. James S. Crossley, Syracuse, N. Y., assignor to Pass & Seymour, Inc., Solvay, N. Y., a Corporation of New York. 957,548.

STARTING DEVICE FOR ELECTRICAL MOTORS. Hans Weichsel, St. Louis, Mo., assignor to Wagner Electric Manufacturing Company, St. Louis, Mo., a Corporation of Missouri. 957,587.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering Societies building, 29 West 39th St., New York. Monthly meetings in New York City.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., W. W. Freeman, Brooklyn, N. Y.; sec., T. C. Martin, 33 West Thirty-ninth St., New York.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railway, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, Oliver building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., L. B. Stillwell; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., Prof. J. D. Hoffman; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, W. S. Cadwell, Chicago, Ill.; sec., Thomas H. Jones, 244 Eighth street, N. E., Washington, D. C. Next convention, Buffalo, N. Y., August 2-5, 1910.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next meeting at Philadelphia, June, 1911.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. N. Lucas; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Omaha, Neb., May, 1911.

INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee, 606 Main St., Peoria, Ill. Next convention, Denver, Colo., September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., G. W. Wright, Baltimore, Md.; sec. and treas., D. L. Gaskill, Greenville, O.

Moments with the Ad. Editor

A department for subscribers edited by the advertising service department of Power and the Engineer.

One of the biggest businesses in the world is that of the General Electric Company.

It is one of those concerns that make our foreign friends talk about the hustling vigor of American enterprises.

And, of course, it is a very large advertiser in technical publications. It might be larger in some—but that's another story!

We recently received a letter from one of our good friends in the Advertising Department in which he says:

"No doubt you will be interested in a comment made by a friend of mine, the engineer in charge of a municipal water and light plant, who is an ardent admirer of POWER. He says: 'I do not like the way POWER advertises its ad. business. You would think it necessary to buy most of the stuff advertised in order to be right, when, as a matter of fact, the most of it can't be used in any one plant.'

"My correspondent bears out my belief in POWER advertising by admitting that he reads the advertising of prominent concerns regularly and finds it interesting and helpful in his business."

Which only proves again that there are two sides to every question.

We have received and heard lots of favorable comment on these advertising editorials from subscribers and advertisers.

And we are glad, too, to get the other viewpoint.

One man, at least, who has been reading these articles on advertising, has received the impression that we advocate it as "necessary to buy most of the stuff advertised" when, as a matter of fact, "most of it can't be used in any one plant."

A surmise very far from what we have tried to convey.

We have consistently advocated reading advertisements, because they are interesting and informative, and buying advertised goods because they are reliable and economical.

But to advocate that a man buy something that he cannot use and does not need—that is like advising a bachelor to lay in a stock of Mellin's Food.

Our contention is that it pays to go through the advertising pages with the same care that is given to the editorial pages.

New inventions, better devices for doing things, improved designs in machinery and equipment, are constantly being brought forward.

And the way to know about them is to read about them.

Certainly the single average plant can find use for but a few of the articles advertised.

But many plants are not using devices that might be used with advantage to the firm and credit to the engineer in charge.

And many of these things will never be heard of by the man who only *skims* the selling section of POWER.

We believe that it *pays* to advertise our advertising—pays readers, advertisers and ourselves.

Advertising is worth knowing about. It's one of the world's great forces. It works days, nights, Sundays and holidays. It makes two blades of wheat grow where but one grew before. It creates business, upbuilds enterprises, spreads information, knits the people of widely separated localities in closer bonds and carries its message in universal language to all peoples in all countries.

All of that is back of the advertisements in your paper.

Focused down to your own problem, every advertisement here has its bearing on power generation and transmission.

For you, it is "wasteless" reading.